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COMPARTMENTAL LUBRICATION SYSTEM.

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Pratt & Whitney Aircraft Group
Government Products Division
United Technologies Corporation
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10 E. M. Beverly

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AIR FORCE AERO PROPULSION LABORATORY
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES
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This technical report has been reviewed and is approved for publication.

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An optimum low vulnerability engine compartmental lubrication system concept was selected from quantitative analysis of five candidate schemes. The five schemes were configured based on qualitative evaluations of many lubrication system components and locations of components. Analysis showed the selected system would provide significant improvements in the areas of reduced vulnerability, maintainability, system cost, frontal area, and increased reliability compared to the F100-PW-100 engine.

The selected compartmental lubrication system was designed, fabricated and tested at the component and full scale system levels. The following significant results from the tests were obtained:

- Demonstrated the durability and performance of a high-speed, 10,000 rpm (2.5 times greater than conventional engine pump speeds) oil pump and drive gear train.
- Deaerated three times the conventional engine air leakage in a small volume oil tank.
- Successfully scavenged (without adverse oil churning) a bearing compartment with increased density due to an oil tank and pump installed within the compartment.
- Successfully demonstrated the Compartmental Lubrication System Concept as an approach to improved system vulnerability for future engine applications.

FOREWORD

This final report was prepared in accordance with Contract No. F33615-75-C-2075, Project No. 3048, Task No. 304806, Work Unit No. 30480681, Development of Compartmental Lubrication System. The contract was conducted under the direction of Mr. L. J. DeBrohun, Project Engineer, SFL of the Air Force Aero Propulsion Laboratory. This report presents the work conducted by Pratt & Whitney Aircraft Group Government Products Division of United Technologies Corporation, P.O. Box 2691, West Palm Beach, Florida, 33402, in accordance with Sequence No. 6 of Attachment 1 (DD Form 1423) of the contract.

The work was performed 6 October 1975 through 1 April 1978 by Pratt & Whitney Aircraft Group under Mr. E. M. Beverly, Program Manager, with Mr. C. E. Swavely providing senior technical and managerial direction. This report was submitted by the author 1 April 1978.

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SUMMARY

The objective of this program was to develop a lubrication system that will reduce turbine engine vulnerability, weight, frontal area and increase reliability. This was accomplished by selecting, through conceptual design studies, a compartmental lubrication system for detailed evaluation, design, and subsequent component and full-scale rig testing.

The F100-PW-100 engine was used as a baseline for sizing components, evaluating performance characteristics, and establishing bearing compartment geometry limitations. An optimum compartmental lubrication system design concept was selected for testing after the trade studies and preliminary design phases were completed. The final system evaluation of the optimum concept was compared with the baseline F100-PW-100 engine with the following results:

- Vulnerability was reduced 28.8 percent
- Maintainability requirements were reduced 5,756 maintenance man-hours per million engine flight hours
- Reliability was increased with 962 fewer part discrepancies per million engine flight hours
- Lubrication system weight was increased 1.7 lb
- Cost was decreased $906 per engine or $4.1 million on a life-cycle basis
- Frontal area was decreased by 80 in.^{2}
- Starting and windmill operation was unchanged
- Time between oil filter changes were decreased approximately 10 percent due to increased air leakage into the No. 1, 4, and 5 compartments resulting from the use of labyrinth mainshaft seals.

The program was conducted in three phases. During Phase I, design trade studies were initiated by formulating a comprehensive list of lubrication system component concepts and possible engine locations. A qualitative evaluation of components and locations was conducted based on previous experience and studies. Five system schemes were configured, using the most promising component concepts and locations. These schemes were configured around the F100-PW-100 engine flowpath and bearing compartment arrangement as a representative engine. A sixth scheme was added to the trade studies to evaluate the armor plating of lubrication system components.

Quantitative analyses were performed on a component basis for each of the schemes and compared to the baseline engine. The armor plate scheme obtained the greatest number of points during the quantitative analysis. However, this scheme was eliminated due to excessive weight (greater than 300 lb). Following the quantitative analyses, an optimum compartmental lubrication system concept was formulated using selected components of the five schemes.

Phase II consisted of (1) a preliminary layout design of the optimum system, (2) re-evaluation of the system as compared to the baseline F100 engine on the basis of

vulnerability, maintainability, reliability, weight, acquisition cost, life-cycle cost, frontal area, starting, windmilling operation, and oil contamination tolerance, and (3) final engine layout design of the optimum system selected.

During Phase III, the optimum compartmental lubrication system design was finalized, fabricated, and tested. A total of 60 hours of run time was accumulated on the critical components through bench tests. This was followed by 67 hours of system tests of which 50 hours was simulated mission endurance time. The system tests provided substantiation of the small, high-speed components integrally mounted with a small volume oil tank in a conventional bearing compartment. The following is a summary of the test results:

- Demonstrated the durability and performance of the 10,000 rpm high-speed (2.5 times conventional engine pump speeds) oil pump and drive gear train

- Deaerated three times the conventional engine air leakage in a small volume oil tank

- Scavenged the bearing compartment preventing adverse heat generation due to oil churning

- Demonstrated the compartmental lubrication system concept as a viable approach to improve system vulnerability for future engine applications.

SECTION I
INTRODUCTION

1. BACKGROUND

The lubrication system is one of the most vulnerable areas in current gas turbine engines. Vulnerability of the lubrication system components to small arms fire and missile shrapnel results primarily from their location on the outside of the engine. A hit in any of the lubrication system components would most likely result in loss of oil to the entire system in a short period of time. With continued operation, this oil loss will lead to bearing and/or gear distress and eventually to loss of the engine.

In the last two decades, significant advances have been made in increasing the thrust/weight ratio of gas turbine engines. This has been achieved primarily through technology improvements of the large engine components, i.e., the compressor, turbine, combustor, and augmentor. Engine lubrication system design refinements and miniaturization have not kept pace with the larger components, primarily because engine program development schedules and funding limitations have precluded investigation of promising lubrication system configurations that incorporate unproven concepts.

Since most of the lubrication system components can be mounted externally to the engine, it has been the tendency to design the lubrication system around the engine rather than making it an integral part of the design requirements. Maintainability considerations have resulted in locating most of the components on the bottom of the engine external to the outer case, increasing their vulnerability.

Lubrication system component state-of-the-art and maintainability have dictated system configuration for current engines. These considerations have resulted in highly vulnerable systems. Recent improvements in engine airframe integration have reduced turnaround time for engine removal and reinstallation in the aircraft to less than 30 minutes. This lessens the importance of lubrication system component exposure on the bottom of the engine as a maintainability criteria. Consideration of lubrication system vulnerable locations, identification of pertinent component state-of-the-art limitations, and appropriate component technology advances can significantly reduce vulnerability with minimum impact on maintainability.

Reduced vulnerability can be achieved by: (1) integrating components within the engine to reduce exposed area, (2) reducing component volume by providing high-speed components, and (3) locating components so that critical items are shielded by engine structure. Locating lubrication components near critical engine components, thereby reducing the overall exposed critical engine/lubrication area, is another means of reducing vulnerability.

Development risk and cost of these integration techniques must reflect a growing concern for reducing overall system costs by maintaining a goal of low-risk development and reasonable production pricing. The ideal system must not adversely impact overall engine performance and weight. Since gas turbine engines must be field maintainable, the lubrication system design implementation must reflect proper considerations for routine engine service and component repair or replacement. Therefore, the problem is not just one of component integration, but component integration in a manner which does not severely sacrifice other important operational criteria.

2. SCOPE

The Compartmental Lubrication System program was a comprehensive design study and experimental program to reduce lubrication system vulnerability, weight, and frontal area, and increase reliability. The finished product is an engine system design with a low vulnerability compartmental lubrication system which was successfully tested at both component and system levels. The compartmental lubrication system program was conducted in three phases, as outlined in the Statement of Work.

Phase I consisted of the quantitative evaluation of five system configurations plus the F100-PW-100 as a baseline engine on the basis of vulnerability, maintainability, reliability, acquisition costs, life-cycle costs, weight, frontal area, manufacturing, assembly, and development considerations, and system compromises. These candidate systems were configured as the result of a detailed qualitative evaluation of various lubrication component concepts and possible engine locations. An optimum concept was selected on this basis for further analysis in the preliminary design effort of Phase II.

The preliminary design of the selected system was divided into three tasks, comprising Phase II of the program. In the first task of this phase the lubrication system components of the selected system were designed into the bearing compartments of the F100-PW-100 engine as a representative engine. In Task II, the selected system was again evaluated using the contract statement of work criteria and compared with the F100-PW-100 lubrication system as a baseline. Task III provided for the refinement and improvement of the advanced system in those areas deemed necessary by the Task II analysis.

Phase III was performed in five tasks consisting of detail design, fabrication, and testing. In Task I, critical components identified in Phase II were detail designed. Task II involved the fabrication of components designed in Task I and testing of those critical components. The remaining components of the advanced system were detail designed in Task III to the extent necessary for experimental evaluation. Rig modifications required for system tests were designed in this task.

Task IV provided for the fabrication of the hardware designed in Task III, and the assembly of the system rig.

Task V successfully demonstrated the advanced system concept through a 50-hour endurance test of the total system conducted on an integrated basis under simulated engine operating conditions.

3. SYSTEM SAFETY ANALYSIS REPORT

The System Safety Analysis conducted during the design phase for component and system testing is documented in Appendix O.

SECTION II
PHASE I — DESIGN TRADE STUDIES

1. APPROACH TO SYSTEM SELECTION

a. General Ground Rules

The F100-PW-100 engine was selected as the baseline engine for comparison with the candidate compartmental lubrication system schemes. During formulation of the schemes, it became apparent that some generalized ground rules would be required to make the results of the analyses applicable to a gas turbine engine in the thrust range of the F100-PW-100 without excessively restricting the study to minor modifications of this engine model. The following ground rules and assumptions were used:

- The engine aerodynamics were not changed from the baseline engine; i.e., the blades and vanes and associated rotating hardware were not modified.

- Static internal structures, such as the bearing compartment walls, were modified as required to accommodate lubrication system components; however, the maximum outer case dimensions were unchanged. Modification to outer case structure to provide access to internal components was acceptable.

- Present F100-PW-100 specifications for fuel temperature at the engine-airframe interface were maintained. This resulted in 200°F maximum fuel temperature at fuel flows of 5000 lb/hr/engine and less.

- Existing maximum fuel and oil temperature guidelines to prevent thermal breakdown were maintained. These limits are 285°F at the fuel control, 325°F fuel nozzle temperature, and 350°F bulk oil temperature out of the engine.

- No deviation from standard gas turbine engine fuels and oils was permitted. MIL-L-7808 or MIL-L-23699 oil was used for the analyses in conjunction with MIL-T-5624 (JP-4 or JP-5) fuel.

- Lubrication system vulnerability was calculated as if the engine was in a test stand, i.e., without reference to a specific aircraft.

- All lubrication system components were sized and selected based on technology advances that could be accomplished with minimum risk during the contract period.

- The engine flight envelope was assumed to be the same as the F100-PW-100.

- Engine lubrication system heat generation was assumed to be the same as that of the F100-PW-100.

- Engine must operate at oil temperatures corresponding to a kinematic viscosity of 13,000 cs (−40°F for MIL-L-23699 and −65°F for MIL-L-7808).

- The statement of work required that the lubrication system design provide an option for internal location of the engine alternator. Review of the engine structure resulted in three candidate locations for the alternator: (1) in front of the No. 1 compartment, (2) in the No. 2-3 compartment, and (3) in the rear of the No. 5 compartment. The No. 5 compartment location was ruled out due to excessive environmental temperatures. The No. 2-3 compartment was ruled out because location of the alternator in this compartment would significantly reduce the space available for other lubrication components. Consequently, the location of the alternator in the front of the No. 1 compartment was selected for all schemes.

Using the above ground rules, a list of all conventional lubrication system components and locations were rated on a qualitative basis. The most promising components and locations were combined to formulate the candidate compartmental lubrication system schemes to be rated against each other and the baseline system on a quantitative basis.

b. Component Identification

During the initial stage of the program, candidate lubrication component concepts and engine locations for these components were identified. This list is shown in Table 1. A qualitative evaluation of these components and locations was made based on experience gained from previous lubrication and accessories studies. The following component concepts were eliminated from consideration on a qualitative basis:

Component/Concept	*Reason for Elimination*
Centrifugal supply or scavenge pump	Not a positive displacement pump; inability to operate with cold oil and any downstream restriction, such as contamination, would result in a reduction in oil flow.
Jet scavenge pump	Same as for centrifugal pump.
Gas turbine drive for pumps	Large volume and weight penalty; hot air in bearing compartments; performance penalty on engine.
Rotating tube centrifugal supply through the shaft from the No. 2-3 to No. 4 compartment.	Requires inner shaft seals at No. 4 compartment; blocks cooling airflow to turbine; results in unvented compressor bore, which would require heavier disk and supports; possibility of coking oil in hot shaft environment; increased balancing problems with shaft.
Vent No. 4 compartment through shaft to No. 2-3 compartment	Same as for preceding concept.
THERMAL SKIN® air-oil coolers in intermediate case struts	Insufficient surface area due to low air side heat transfer coefficients. Difficult to remove for inspection or repair.

The remaining component concepts were combined into five low vulnerability lubrication schemes, which were rated against each other on a quantitative basis. A sixth scheme was added to evaluate armor plating of lubrication system components; however, this scheme was eliminated due to excessive weight (greater than 300 lb).

TABLE 1
LUBRICATION SYSTEM COMPONENT CONCEPTS AND ENGINE LOCATION

	Possible Locations of Engine Lubrication System Components					
Lubrication System Components	*Bearing Compartments*	*Engine Inner Wall*	*Struts and Vanes*	*Bypass Ducts*	*External of Engine*	*Fore and Aft End Compartments*
Oil Supply and Scavenge Pumps						
Gear	X				X	X
Vane	X					
Centrifugal	X					
Jet	X				X	
Rotating Tube	X					
Blowdown Scavenge System	X					
Pump Drive Systems						
Geared through Tower Shaft	X				X	
Geared-off Rotor	X					X
Filter						
Bypass	X				X	X
Nonbypass	X				X	X
Centrifugal					X	
Heat Exchangers (Coolers)						
Plate Fin				X		
Shell and Tube					X	
Finned Wall		X				
THERMAL SKIN*			X			
Heat Pipes	X				X	
Deaerators - Deoilers						
Centrifugal					X	
Can	X				X	X
Rotor	X					
Oil Tanks						
Internal Reservoir	X					X
External Reservoir					X	
Integrated	X					X
Breather						
Scavenge	X				X	
Vent Tube	X				X	
Chip Detectors						
Magnetic	X				X	X
Bypass Valves						
Filter	X				X	X
Cooler	X				X	X

c. Quantitative Analysis Criteria — Phase I

The most promising component concepts were combined to create five candidate compartmental lubrication system schemes. Each of these was rated quantitatively according to the weighted criteria listed in Table 2. The weighted values of this table were coordinated with the AFAPL Project Engineer. Each of these schemes was evaluated on a differential value basis (i.e., Δweight, Δcost, etc.) as compared to the baseline engine. The quantitative evaluations were made using mechanical layout studies, with component sizes substantiated by numerical analyses.

TABLE 2
WEIGHTED CRITERIA COORDINATED WITH AFAPL

Criteria	*Maximum Point Allotment*	*Comparison to Best Scheme Factor**	*Rating*
Vulnerable Area Reduction	30		
Maintainability	25		
Reliability	10		
Acquisition Costs	5		
Life Cycle Costs	5		
Weight	10		
Frontal Area	8		
Manufacturing, Assembly, and Development Considerations	3		
System Compromises	4		
Σ	=100		

Rating — The best system in a given criteria received maximum point allotment (weighting factor) assigned to that criteria. Other schemes received points on a comparative basis with the best scheme.

Rating = (*Maximum Point* Allotment) $\times$ (Comparison to Best Scheme Factor).

* Comparison to best scheme factor = 1.0 for best scheme and is proportioned to each lower rated scheme directly depending upon its relationship to the best scheme. For example, if the best scheme weight is 300 lb, while another scheme has a weight of 600 lb, its comparison-to-best scheme factor is 300/600 = 0.50.

d. Methods of Quantitative Analysis for Each Rating Criteria — Phase I

(1) Vulnerability

Vulnerability was quantified by comparing vulnerable areas. The procedures followed and the assumptions used for this analysis were:

(a) Six views were used that were considered vulnerable as a projectile target. They were the front, rear, top, bottom, and left and right sides.

(b) Each scheme was separated into various components, and vulnerable area (VA) was calculated for each component in each view. Only those components in the oil system that changed in size and/or location were included in the analysis:

- No. 1, 2-3, 4, and 5 Bearing Compartments
- Oil Tank
- Oil Pumps (Boost and Scavenge)
- Oil Filter
- Fuel/Oil Coolers
- Air/Oil Coolers
- Main Gearbox
- Plumbing (Oil System Only)

(c) The vulnerable area was determined by multiplying the component projected area by its kill probability. The kill probability is the probability that the engine will fail to deliver flight sustaining power if the component is hit. The kill probabilities are experience factors based on test data from previous engines.

(d) The vulnerable areas were calculated for A kills (loss of flight sustaining power in 5 min) and B kills (loss within 30 min) for 30- and 50-caliber armor piercing projectiles traveling at 1500 and 2500 ft/sec.

(e) An "A" kill is defined as a hit to the fuel system or to the main fuel pump drive train resulting in a loss of gas generator fuel flow. Also, a critical hit to a main rotor bearing results in a loss of rotor support and then loss of power within 5 minutes.

A "B" kill is defined as a hit to the oil system resulting in a loss of oil pressure to the main bearings and subsequent rotor seizure. Table 3 shows the various components with failure modes and minimum size and speed of projectiles necessary to cause the respective kills.

(f) The projected area for oil system plumbing was established from a previous engine fluids study performed for the F100-PW-100 engine. The various schemes were calculated as some percentage of the baseline projected area for each of the six views. The vulnerable area was then computed using these estimated projected areas for each scheme.

(g) The vulnerable area of the scheme for any view is the sum of the component vulnerable areas in that view for each type kill, speed, and size projectile. The vulnerable areas for each scheme were tabulated in terms of a difference from the baseline in square inches. The lowest numbers showed the scheme that was least vulnerable in each view for each category.

TABLE 3
COMPONENT MALFUNCTION MODES

Part	Malfunction Mode	Kill	Minimum Required to Cause Kill: Size, Cal	Speed, ft/sec
Main Rotor Bearing	Bearings shatter when hit, resulting in loss of rotor support	A	30 50	2000 1000
Towershaft and Other Drive Shafts for MFP	Hit on referenced parts results in loss of main fuel pump power supply, causing loss of engine fuel supply	A	50	2000
Bullgear in 2-3 Bearing Compartment	Hit on referenced part results in loss of main fuel pump power supply, causing loss of engine fuel supply	A	50	2000
Bevel Gear in 2-3 Bearing Compartment and Gears in Main Gearbox	Hit on referenced parts results in loss of main fuel pump power supply, causing loss of engine fuel supply	A	50	1000
Bearings for MFP Drive Shafts	Hit on referenced parts results in loss of main fuel pump power supply, causing loss of engine fuel supply	A	30 50	1500 1000
Fuel/Oil Coolers	Hit results in loss of Gas Generator fuel flow	A	30	500
Oil Tanks, Filter, Air/Oil Coolers, Oil Plumbing	Hit causes loss of oil, resulting in seizure of rotors	B	30	500
Oil Pumps, Main Gearbox, Bearing Compartments	Hit causes loss of oil, resulting in seizure of rotors.	B	30	1000

(h) The six different views were weighted to establish a criteria for comparing the vulnerable area in each view as follows:

Views	Weights, %
Front	5
Rear	15
Top	10
Bottom	30
Left Side	20
Right Side	20

The bottom was considered most vulnerable due to the likelihood of heavy ground fire. Likewise, the front view was least vulnerable due to the relatively small chance of head-on fire from enemy aircraft.

(i) The "A" and "B" kills for ballistic speed and projectile size were then individually averaged and weighted for each view. The views were then added together for each scheme whereby a best scheme was determined for each type kill. Assuming the hit probability of each kill to be equal, the two kills were then averaged together to determine the scheme that was least vulnerable overall. This scheme obtained a comparison to best scheme factor of 1.0 and the full 30-point vulnerability allotment. Less effective schemes received a percentage of this rating based on their relative vulnerable areas.

(2) Maintainability

The basis for the measurement of maintainability is maintenance man-hours per engine flight hour (MMH/EFH). Maintenance man-hours are estimated task times required to remove and replace all components within the engine. Estimates were made using the "Standards for Maintenance Time Estimates for Part Replacement" or by actual measurement of specific tasks performed during engine assembly or disassembly. The maintenance task time for each component was multiplied by its parts failure and discrepancy rate to determine its MMH/EFH. The parts failure and discrepancy rates were obtained from our reliability prediction model.

For this study, task times are expressed as a difference in MMH from the baseline for each component or module that requires some change in maintainability. This means that to remove/replace a pump located with the No. 2-3 compartment, for example, there is a much greater MMH number than baseline because the inlet fan module must be removed to enter the No. 2-3 bearing compartment and gain access to the pumps. Likewise, there is a different parts discrepancy rate from baseline for some components because of their location and environment.

Since the ΔMMH/EFH for all schemes were small in comparison to the absolute total engine values, it was decided to deviate from the previously stated method of calculating the comparison to best scheme factor as a ratio of absolute values. This method would not give a large spread in maintainability rating points and adequately distinguish the advantages of one scheme over another. The method used for this analysis was to determine the ratio of the range of ΔMMH/EFH values minus the ΔMMH/EFH value for the given scheme, divided by the range of ΔMMH/EFH. This factor times the maximum point allotment provided the scheme rating.

(3) Reliability

The basis for the measurement of reliability used in this study was part failures and discrepancies, expressed as discrepancy rates. The discrepancy rates were obtained from the reliability prediction mathematical model and reflect the number of discrepancies expected to occur after the engine has reached maturity. An engine design is considered mature after it has accumulated approximately one million engine flight hours.

To determine the overall reliability rate for each scheme, a discrepancy rate for each major component was predicted and the rates summed to obtain the total discrepancy rate for that scheme.

As with the maintainability analysis, it was necessary to modify the procedure for calculating the comparison to best scheme factor to adequately distinguish the advantages of one scheme over another. The method used was to determine the ratio of the difference between the worst scheme Δreliability values and the given scheme Δreliability values divided by the absolute difference in the worst and best scheme Δreliability values. This factor times the maximum point allotment provided the scheme rating.

(4) Acquisition Costs

Cost estimates for this analysis were made using methods in general use by P&WA for many years, based on a standard cost accounting system. Extensive cross-reference files of vendor and in-house manufacturing information are maintained for detailed component analysis. This comprehensive estimating data base facilitates accurate cost forecasting. The scheme with the lowest acquisition cost was assigned a comparison to best scheme factor of one. All other schemes were rated against the best scheme proportional to their total lubrication system cost.

(5) Life Cycle Costs

A life cycle cost comparison was made of the five schemes being evaluated, based on an air superiority fighter application having 15-year life cycle. This study considered the differences in acquisition, operating, and support costs for 1000 engines during peacetime operations. Savings due to lower combat attrition rates, resulting from decreased engine vulnerability, were not included in this comparison since the vulnerability criterion received a separate, heavily-weighted point allotment in the weighted criteria rating system.

Ground rules and assumptions used in the life cycle cost comparison of the six candidate schemes were:

- 1000 total engines, including 15 percent uninstalled spares
- 75 percent of the installed engines operational, flying 25 hr per month for 15 years
- Base labor rate = $16.25 per maintenance man-hour (MMH); depot labor rate = $23.24 per MMH.

Acquisition costs included only those associated with changes in engine configuration, since detailed airframe installation differences were not defined during this study. Since the compartmental lubrication system would be incorporated as part of a completely new engine, development cost differences between the schemes were assumed to be negligible and were excluded from the comparison. Operating and support cost differences fall into the following categories:

- Maintenance Labor — Based on changes from the baseline engine in maintenance tasks times and frequencies
- Recurring Spare Parts — Based on differences in production cost, usage, and repairability
- Fuel and Oil Costs — Considered the same for this study, since all schemes have the same inherent fuel and oil consumption as the baseline engine.

(6) Weight

The weight analysis for this study was conducted by comparing each configuration to the F100-PW-100 Bill-of-Material components. Each configuration was weighed from layout drawings, where thickness and material assumptions were made for most components. Items that were similar to existing hardware were estimated by weighing the discrete differences and applying the resultant delta to the overall difference of the configurations. Hardware for each configuration was then grouped by function to isolate areas of significant weight difference and

to provide a method of rating each scheme to each other and to the Bill-of-Material design. The scheme with the lowest weight was then assigned a comparison to best scheme factor of one and all other schemes were rated against the best scheme.

(7) Frontal Area

A frontal area comparison was made by calculating the projected frontal area of the entire engine, including the core and all accessories for each of the six schemes and the baseline engine. The augmentor nozzle was not included as part of the projected frontal area, since a hit on this component would not result in a loss of engine fluids or a malfunction of the rotating machinery. The scheme with the minimum frontal area was then assigned a comparison to best scheme factor of one.

(8) Manufacturing, Assembly, and Development Considerations

This comparison was made by first listing the manufacturing, assembly, and development difficulties that must be considered for each scheme. Examples of these difficulties are if a component is difficult to assemble, requires stringent tolerances, or will require extensive development. Each scheme was compared with the baseline scheme for determining the magnitude of the problem. However, where the baseline scheme was more complicated than any of the other schemes, it was rated accordingly. Each of the manufacturing, assembly, and development difficulties was rated from -1 to -10, based on the severity of the problem, with the most severe problem getting a rating of -10. The points for each scheme were then totaled, and the scheme with the minimum absolute value of points was assigned a comparison to best scheme factor of one. Any other scheme received a fractional value for this factor, obtained by dividing the absolute value of points for the best scheme by the absolute value of points for that scheme.

(9) System Compromises

This comparison was made by first listing the modifications and compromises made to incorporate each scheme, i.e., relocate bearing support, which will reduce critical speed margin, increase number of service ports, decrease the accessibility of components, etc. The severity of each compromise was then rated from -1 to -10, with the most severe compromise receiving a rating of -10. The points for each scheme were then totaled, and the scheme with the minimum absolute value of points was assigned a comparison to best scheme factor of one. All other schemes received a fractional value for this factor based on a numerical ratio of the absolute value of total points compared to the best scheme.

2. Compartmental Lubrication System Scheme Definition

A definition of the various lubrication schemes evaluated during Phase I studies is presented in this section in three parts. The first, component arrangement, describes the location of the lubrication system components. Part two, system flowpath, traces the lubrication oil around its entire flow circuit from oil tank to compartment and back. A discussion of the various analytical and mechanical design considerations pertinent to each lubrication scheme is presented in the third part. This includes assumptions used to facilitate analysis and technical approach to problem solving.

A summary of lubrication system components size for each of the advanced schemes is presented in Appendix A.

a. Candidate Scheme I

(1) Component Arrangement

This lubrication system concept used the No. 2-3 bearing compartment to house a majority of the lubrication system components, as shown in Figure 1. The main oil supply pump, No. 2-3 and 4 scavenge oil pumps, oil tank, can deaerator, oil filter, and breather system are all located within the No. 2-3 compartment to reduce vulnerability. The alternator is located in the No. 1 bearing compartment and is driven directly off the low rotor. The No. 1 scavenge pump, mounted adjacent to the alternator, is driven by a gear-driven train integral with the alternator shaft drive. The No. 5 scavenge pump is located in the No. 5 compartment and gear driven off the low rotor.

The gearbox is mounted on top of the engine and coupled with a towershaft, which is run through an adjacent support strut in the No. 2-3 compartment. The deoiler and breather pressurizing valve are gearbox mounted.

The air/oil coolers are located in the fan duct consistent with the baseline (F100-PW-100) system. The fuel/oil coolers (F100-PW-100 baseline) and all the external lines are located on top of the engine. All of the scavenge return lines incorporated chip detectors.

A dipstick is used for determining oil level in the oil tank during servicing.

(2) System Flowpath

Oil is supplied from the oil tank to the main oil pump, then passes through an oil filter before entering the cooling system outside the compartment. The main oil pump, oil filter, and oil coolers are protected from cold starts (and a plugged filter) by bypass circuits activated by pressure relief valves. The oil flow is then split into separate paths for the No. 1, 2-3, 4 and 5 bearing compartments and gearbox. A boost oil pump is not required because the No. 4 compartment utilizes a breather line for removing the air leakages, thus preventing significant compartment pressure levels.

The No. 1 and 5 compartments are capped and use scavenge pumps, located inside their respective compartments, to transfer the compartmental air leakages and oil flow back to the can deaerator, located in the oil tank. The No. 4 compartment air leakages are breathed directly back to the gearbox, while the oil is scavenged back to the can deaerator by a scavenge pump, located adjacent to the oil tank. The oil in the gearbox is gravity-drained down the towershaft strut, where it is picked up along with No. 2-3 compartmental oil by a scavenge pump feeding from the oil sump on the bottom side of the oil tank. The air, separated from the oil in the oil tank, is breathed back to the gearbox through the breather line, where it combines with the No. 4 compartment air leakage prior to venting overboard through the deoiler and breather pressurization valve.

(3) Design Considerations

The intent of this lubrication system configuration is to provide reduced vulnerability by using the individual bearing compartments to house critical lubrication system components.

To achieve this objective, it was necessary to reduce the size of the lubrication pump. This permitted the placement of the No. 1 and 5 scavenge pumps into their respective compartments. Mounting and driving the remaining pump elements within the No. 2-3 compartment required the redesign of the No. 2-3 bearing support to facilitate the pumps and provide maximum oil tank capacity. The lubrication pump size was reduced by the following procedure:

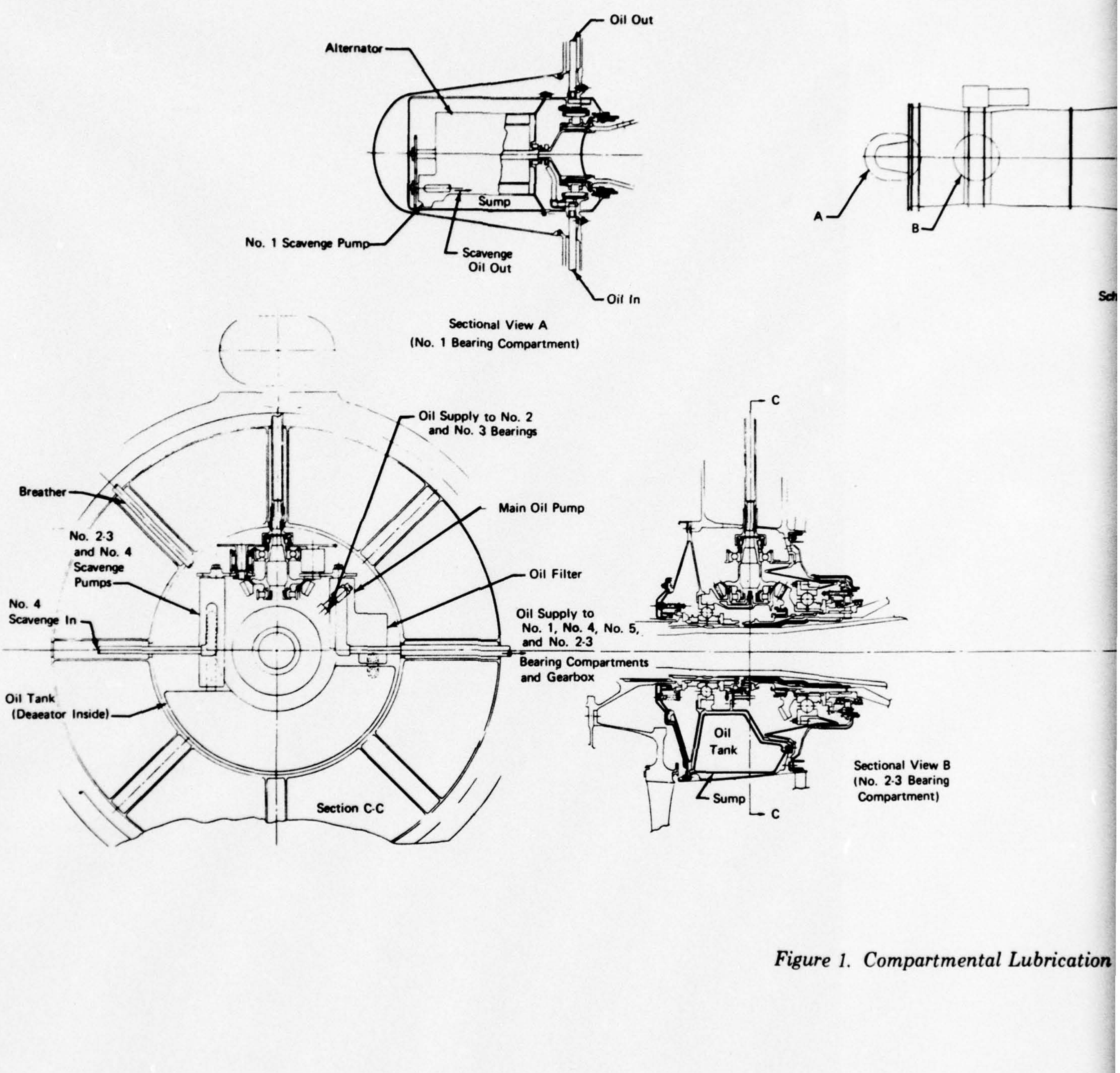
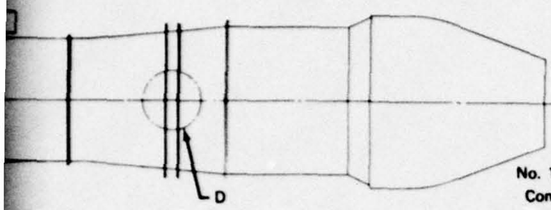
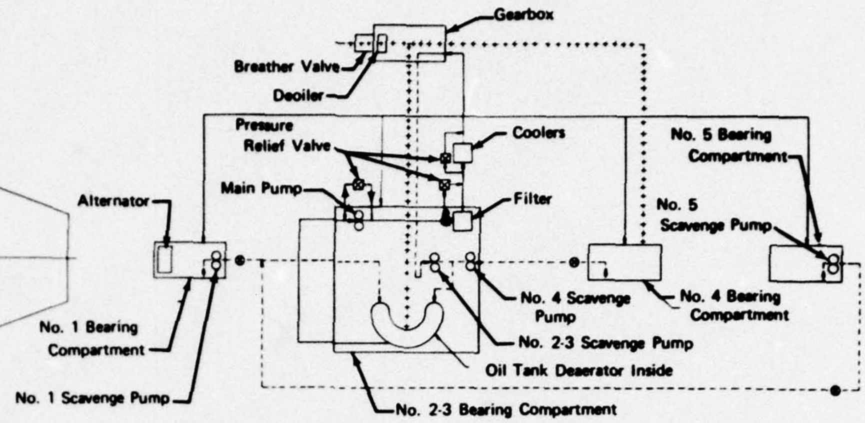


Figure 1. Compartmental Lubrication

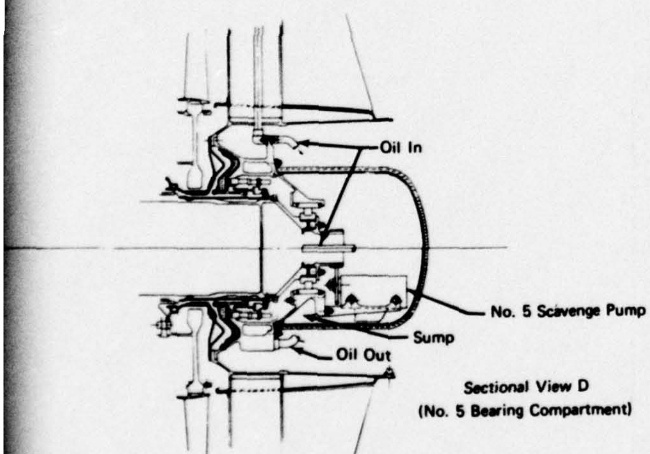


Scheme I



Lubrication System Schematic

Legend:
——— Supply Line
– – – Scavenge Line
+ + + Breather Line
● Chip Detector



Sectional View D
(No. 5 Bearing Compartment)

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ubrication System — Scheme I

2

- Pump elements were scaled from an ST9 high-speed gear pump, which has a designed speed of 10,000 rpm. (Baseline F100-PW-100 lubrication design speed is 4000 rpm.)

- The boost pump element was eliminated, and the No. 4 scavenge element size was reduced by using a breather line for the No. 4 bearing compartment for venting air leakages to the gearbox.

This permitted the oil supply and No. 2-3 and 4 scavenge pumps in the No. 2-3 bearing compartment, along with an oil tank of 1.82-gal maximum capacity. The oil tank capacity was considered insufficient to provide adequate make-up oil for mission requirements, and to prevent low and fluctuating oil pressures. A supplemental oil tank, externally mounted, would have been evaluated if this scheme had been selected for further studies.

Vulnerability was further reduced by locating the alternator in the No. 1 compartment, similar to the arrangement previously demonstrated successfully under Contract N00140-73-C-0126, which used the forward compartment of the J52 engine. Locating the alternator in the No. 1 compartment and driving it directly off the low rotor provides the following benefits:

- Vulnerability is reduced, since the bearing compartment walls shielded the alternator.

- Driving directly off the low rotor eliminated a gear set in the gearbox.

- A mainshaft seal was added, canceling the seal eliminated in the gearbox; however, the mainshaft application provided better accessibility for cooling provisions.

- The alternator used to drive the No. 1 scavenge pump provided a convenient arrangement.

This alternator location/drive configuration provided the following areas of concern:

- Low-rotor drive during engine start may not provide sufficient electric output to meet control system requirements.

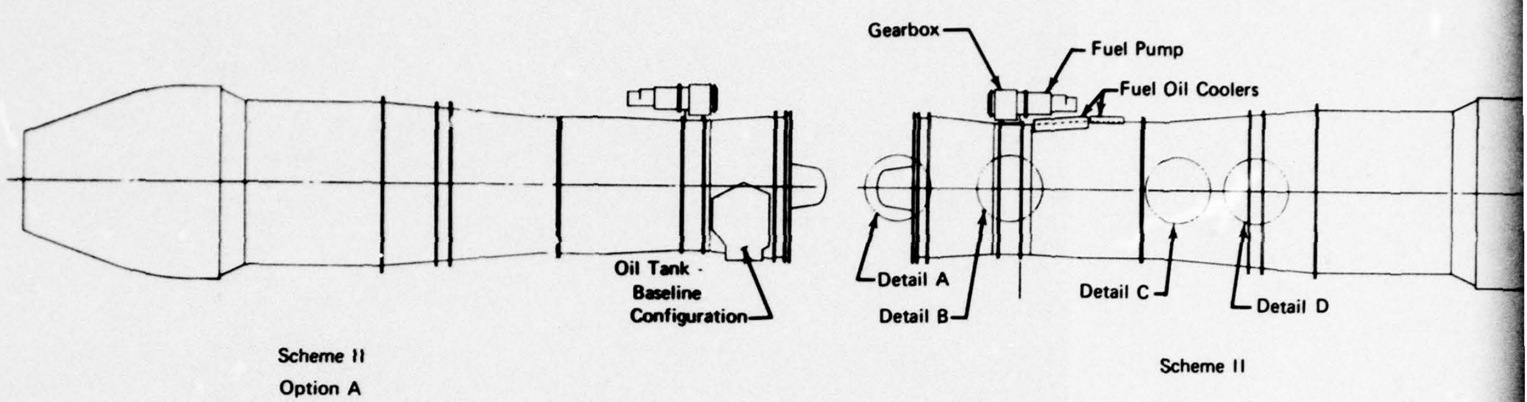
- Mounting the alternator on the low rotor impacts the rotor dynamics, which adversely influence critical speed margin.

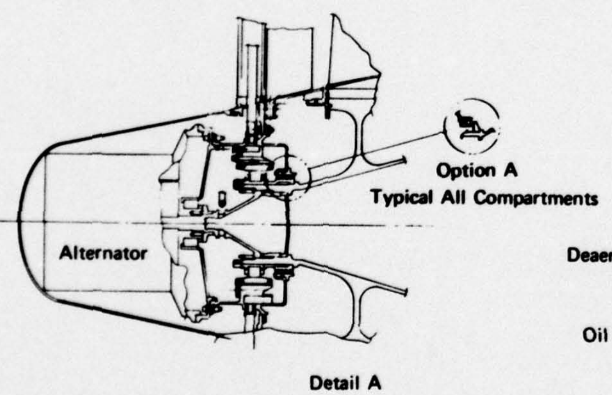
Locating the No. 5 scavenge pump in the No. 5 bearing compartment required rearrangement of the baseline design. The No. 5 bearing was moved aft of the baseline position to provide for a drive gear off the low-rotor shaft to drive the scavenge pump. The increased rotor length resulted in an estimated 5 percent reduction in shaft critical speed margin.

b. Candidate Scheme II

(1) Component Arrangement

This lubrication system scheme, in similar fashion to Scheme I attempts to reduce vulnerability by using the No. 2-3 bearing compartment to locate major lubrication components, as illustrated in Figure 2. The main oil supply pump, No. 2-3 scavenge oil pump, oil tank, can deaerator, oil filter, and breather system were located within the No. 2-3 compartment. The alternator is located in the No. 1 bearing compartment and driven directly off the low rotor.





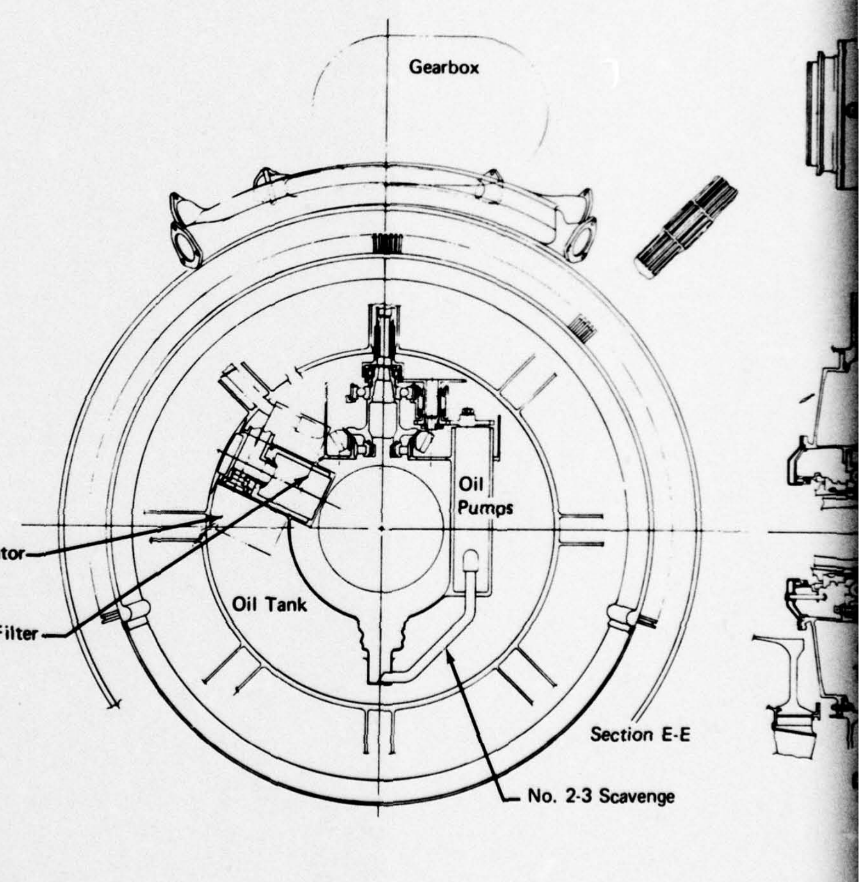


Figure 2. Compartmental Lubrication Syst

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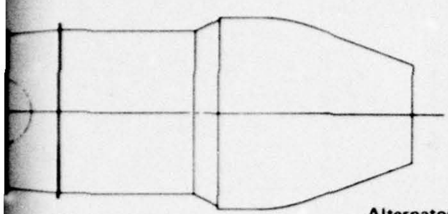
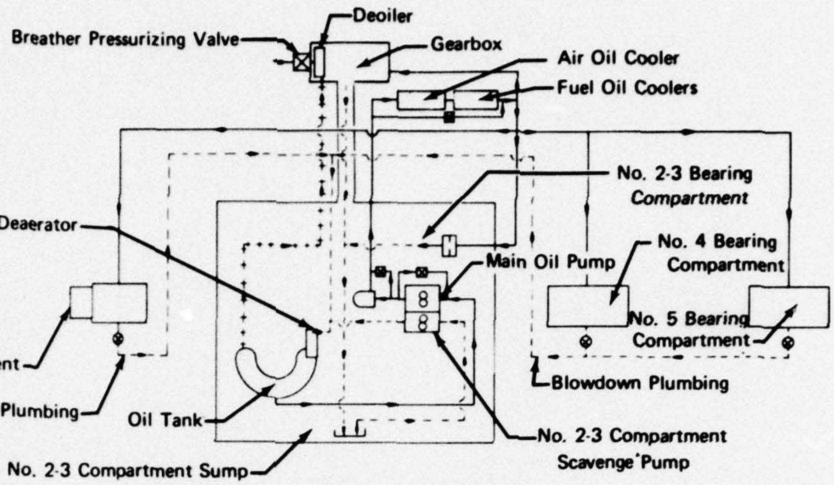
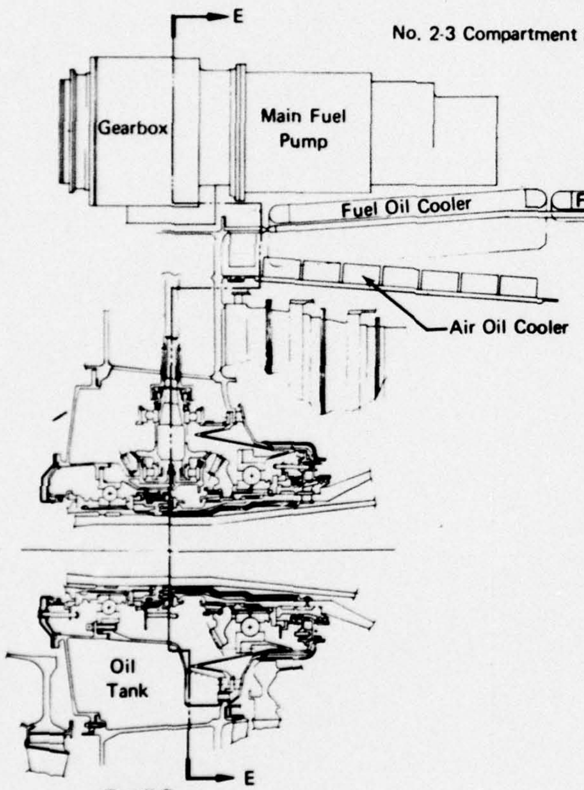


etail D

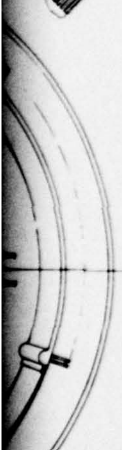


Flow Schematic
for
Scheme II

Legend:
——— Pressure Line
– – – Scavenge Line
+ + + Breather Line
■ Check Valve
⊕ Chip Detector

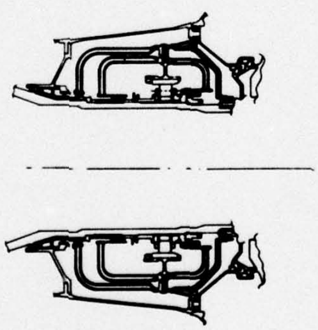


Detail B

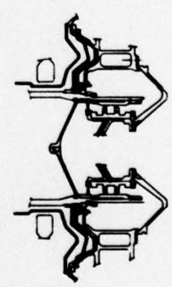


ection E-E

Scavenge



Detail C



Detail D

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ntal Lubrication System — Scheme II

2

The gearbox is mounted on top of the engine and driven by a towershaft running through a vertical support strut in the No. 2-3 compartment. The deoiler and breather pressurizing valve are gearbox mounted.

Blowdown plumbing lines for scavenging the No. 1, 4, and 5 compartments are located on top of the engine and incorporate chip detectors.

A finned wall air/oil cooler is located in the inner duct fairing, and plate-fin fuel/oil coolers are located in the fan duct wall.

A dipstick is used for determining oil level in the oil tank during servicing.

(2) System Flowpath

Oil is supplied from the oil tank to the main oil pump, then passed through an oil filter before entering the cooling system outside the compartment. The main oil pump, oil filter, and oil coolers are protected from cold oil starts (and a plugged filter) by bypass circuits, activated by pressure relief valves. Upon exiting the fuel/oil coolers, the oil flow is split into separate paths to the gearbox and No. 1, 2-3, 4, and 5 bearing compartments. Gearbox oil is gravity-drained down the towershaft support strut to the No. 2-3 compartment sump. A scavenge pump transfers gearbox and No. 2-3 compartment oil from the sump to the can deaerator. External blowdown lines provide scavenging for the No. 1, 4, and 5 bearing compartments. The blowdown plumbing lines are sized to maintain low compartment pressure levels, eliminating the requirement for a boost oil pump.

The air separated from the oil by the deaerator in the oil tank is breathed back to the gearbox through the breather line. Prior to venting overboard through the breather pressurizing valve, the breather flow passes through the deoiler, which removes the remaining oil vapor from the air.

(3) Design Considerations

This scheme attempted to further the vulnerability goals discussed in Scheme I by eliminating the No. 1, 4, and 5 scavenge pumps. Bearing compartment scavenging is achieved through the use of blowdown lines. The maximum oil tank capacity of this scheme is increased 40 percent over Scheme I (to 2.5 gal), primarily by using the bearing support structure and intermediate case to configure the oil tank. Unlike the oil tank configuration of Scheme I, which used a separate sheet metal enclosure, this tank design did not completely seal the oil cavity from the No. 2-3 bearing compartment. At specific engine attitudes, the tank oil could enter and flood the bearing compartment. This tank design approach was considered an improvement over Scheme I which was too small to meet performance requirements. Even by eliminating the No. 4 scavenge pump and using the compartment boundaries for tank walls, the 2.5-gal capacity was still marginal in meeting performance.

The blowdown line sizes (1.0 in. OD) are sufficiently large to ensure low compartment pressures and prevent possible compartment oil loss during engine deceleration. Mainshaft carbon face seals (baseline) are eliminated in the No. 1, 4, and 5 compartments and replaced with labyrinth seals. This is to provide the air leakage required to adequately scavenge the compartments of oil.

Access to the oil filter was through an access plate in the OD of the intermediate case as shown in Figure 3. Upon removal of the coverplate and filter housing fasteners, a threaded tool is attached to the threaded boss on top of the filter housing. This provides for removing the filter assembly. Once outside the engine, the filter element can be easily removed from the filter housing for cleaning or replacement. The filter assembly can be reinstalled within the engine in a reverse manner to that previously described.

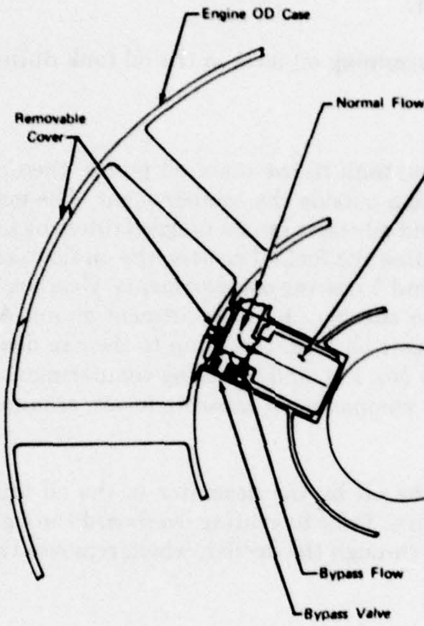

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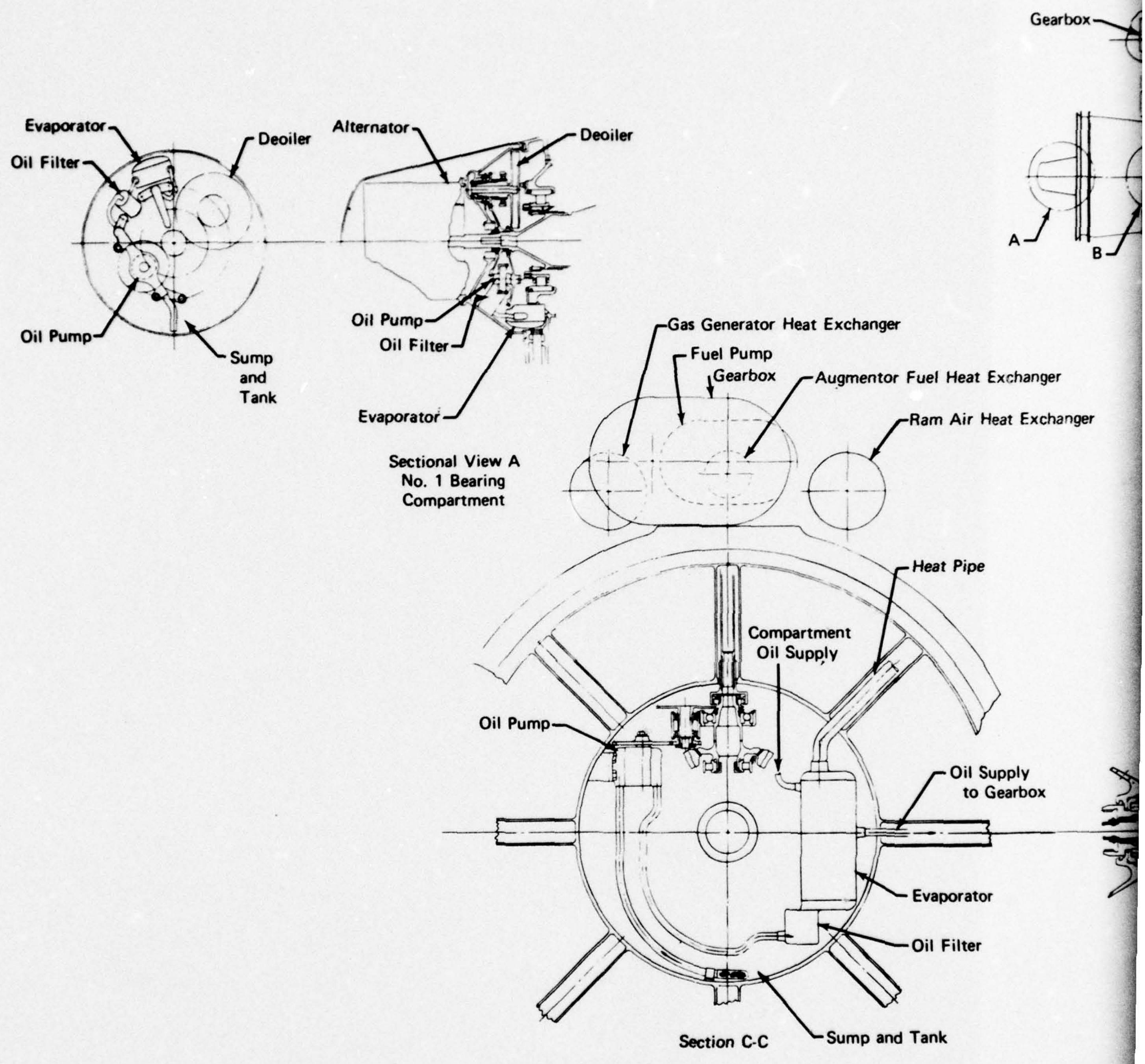
Figure 3. Access to Internal Filter Through Coverplates

c. Candidate Scheme III

(1) Component Arrangement

This scheme utilized individual, self-contained lubrication systems for each bearing compartment as shown in Figure 4. Each bearing compartment contains an oil sump, supply pump, evaporator, heat pipe, deoiler, oil filter, and breather line. Oil bypass flowpaths are provided around the pumps, filters, and evaporators to account for cold oil starts and plugged filters. No external oil plumbing lines are required since the oil never leaves any of the bearing compartments.

Each supply pump was provided an individual bypass valve for cold oil starts and an *integral* filter and chip detector which were accessible through coverplates or probe holes in the outer cases.


Gearbox
Evaporator
Oil Filter
Deoiler
Alternator
Deoiler
A
B
Oil Pump
Sump and Tank
Oil Pump
Oil Filter
Evaporator
Sectional View A
No. 1 Bearing Compartment
Gas Generator Heat Exchanger
Fuel Pump Gearbox
Augmentor Fuel Heat Exchanger
Ram Air Heat Exchanger
Heat Pipe
Compartment Oil Supply
Oil Pump
Oil Supply to Gearbox
Evaporator
Oil Filter
Sump and Tank
Section C-C

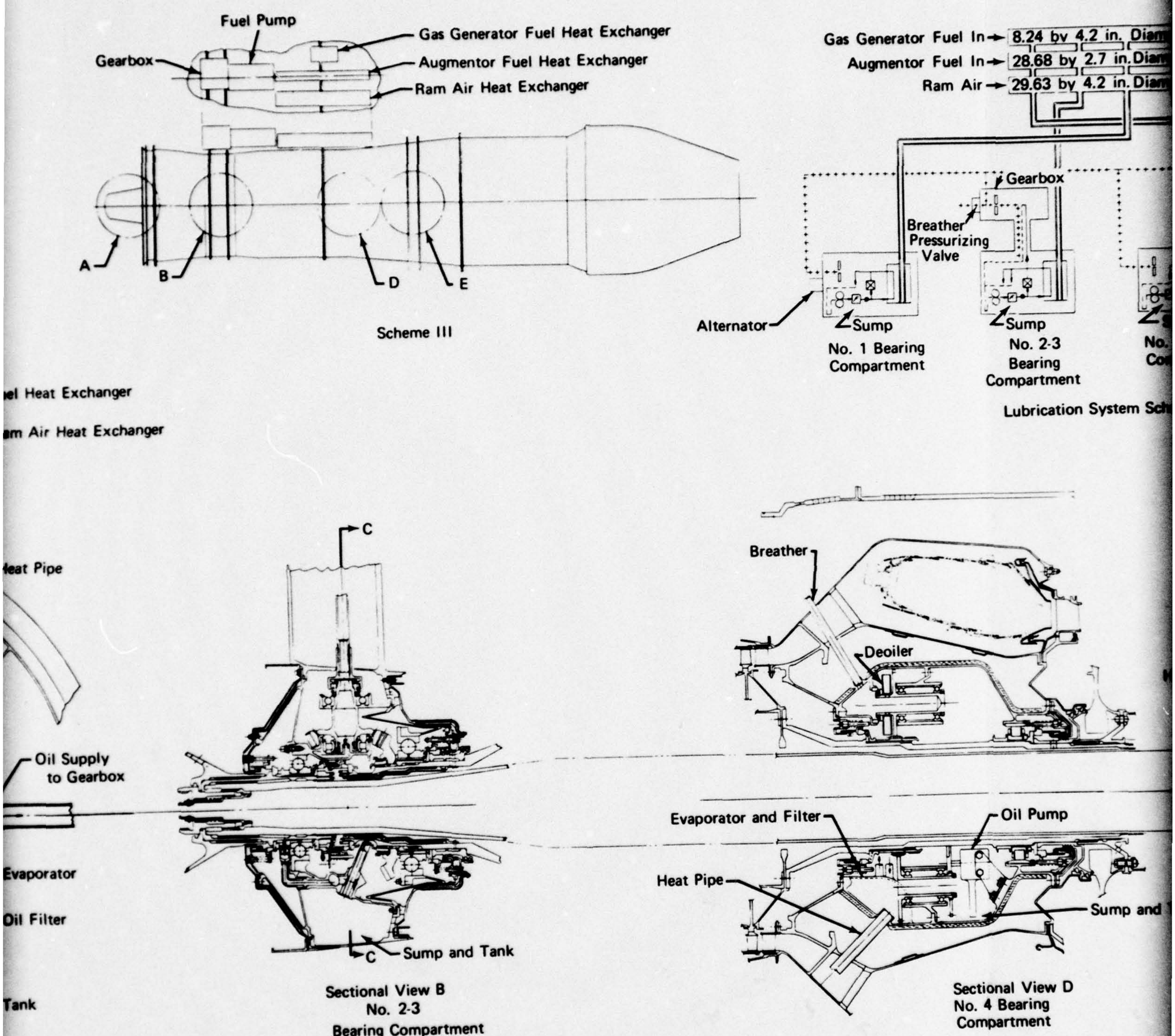
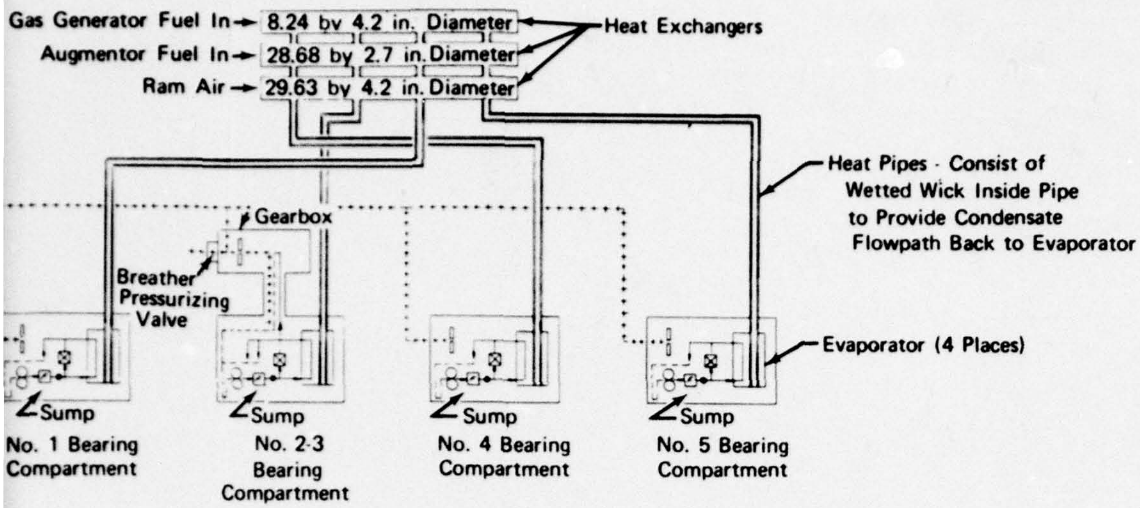
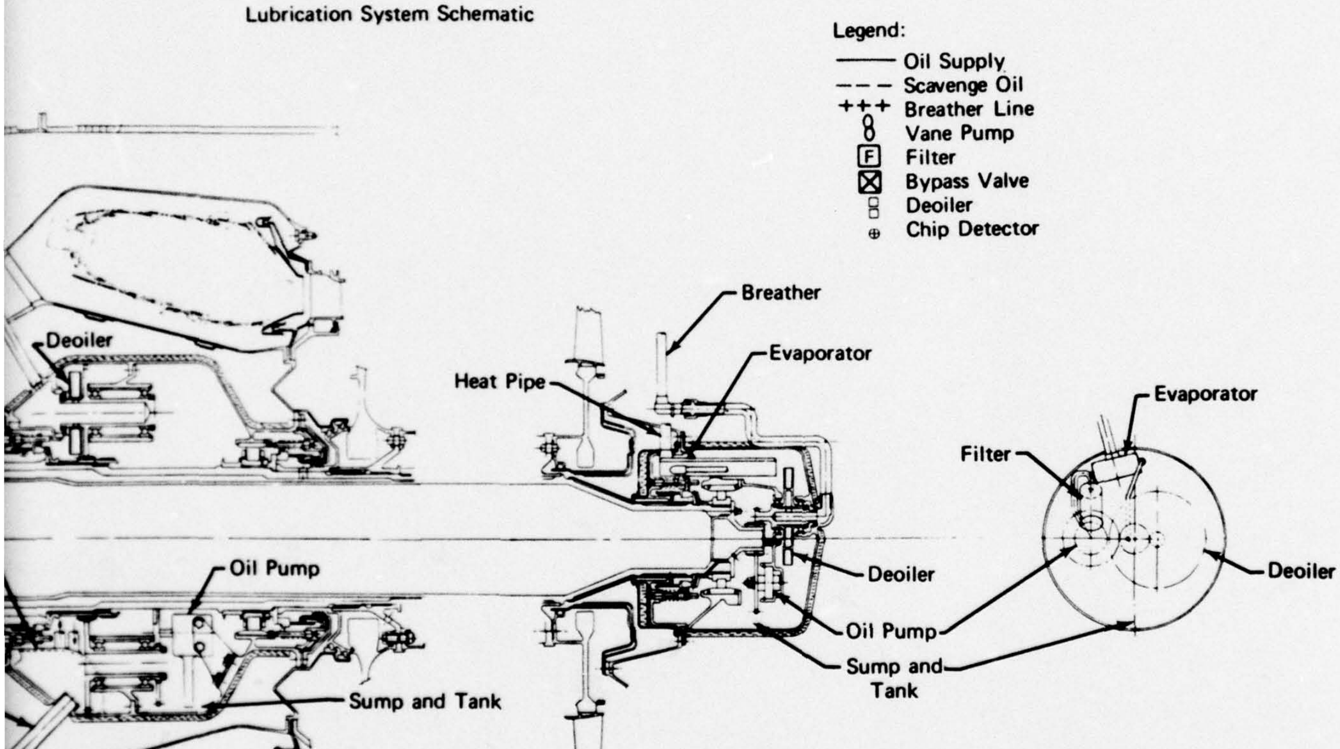


Figure 4. Compartmental Lubrication System — Scheme III



Lubrication System Schematic



Sectional View D
No. 4 Bearing
Compartment

Sectional View E
No. 5 Bearing
Compartment

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3

The heat pipes were extended outside the respective bearing compartments and were structurally integral with the fuel and air coolers on top of the engine. Multiple heat pipes could have been used for each compartment to reduce vulnerability and improve survivability, but this would have greatly complicated the system.

The gearbox is mounted on top of the engine and driven by a towershaft running off the high rotor through a support strut in the No. 2-3 bearing compartment. The individual breather lines are joined together for overboard venting through the gearbox-mounted breather pressurizing valve.

Individual dipsticks are used in each bearing compartment to determine oil level during servicing.

(2) System Flowpath

Oil is supplied to each oil pump from the compartment oil sump and passed through an oil filter and then through the heat pipe boiler prior to supplying the compartment requirements. The gearbox oil is gravity drained down the towershaft strut to the No. 2-3 compartment. This flowpath is identical in each compartment. Each compartment uses a deoiler, located at its breather pipe inlet, for separating oil from the air and venting compartmental air leakages. All of the compartment breather lines are combined externally to a single line which routes the air leakage overboard through the breather pressurizing valve mounted on the gearbox. The heat in the oil is transferred to an intermediate media (water) in the heat pipe. Air and fuel condensers mounted on top of the engine are integral with the heat pipes and transfer the heat of water condensation to the fuel and air in these external coolers.

(3) Design Considerations

Reduced vulnerability is achieved in this scheme by using individual, self-contained lubrication systems within each bearing compartment. This eliminates all external oil supply and scavenge lines and locates all critical lubrication system components, such as pumps, oil sumps, filters, deoilers, and oil cooling devices within the confines of each bearing compartment. The alternator is located within the No. 1 compartment similar to the preceding schemes. No scavenge pumps were required; oil is gravity drained to a sump on the bottom of each compartment which supplies the compartment supply pump. Each compartment has its own deoiler and breather line. The key to a self-contained oil system is the cooling technique. The oil is pumped through a filter to a heat pipe boiler or evaporator which resembles a tube-shell heat exchanger. The oil circulates across tube bundles transferring heat into the intermediate media in the tube or heat pipe. This heating process causes the intermediate media to boil resulting in the vapor traveling from the evaporator to ram air and fuel coolers located outside the bearing compartments on top of the engine. These external coolers condense the vapor from the heat pipes, with the resulting liquid returned to the evaporator by capillary wicks, which lined the heat pipe. This continuous process releases a steady flow of heat which is transferred from the oil to the external condensers through the intermediate media in the heat pipe. A typical heat pipe boiler is shown schematically in Figure 5.

A schematic illustrating a complete evaporator and condenser system is shown in Figure 6. The actual system uses three condensing elements: a ram air cooler, an augmentor fuel cooler, and a gas generator fuel cooler. Figure 7 depicts a schematic that illustrates the entire vapor/wick flowpath through all the condensing elements of this scheme.

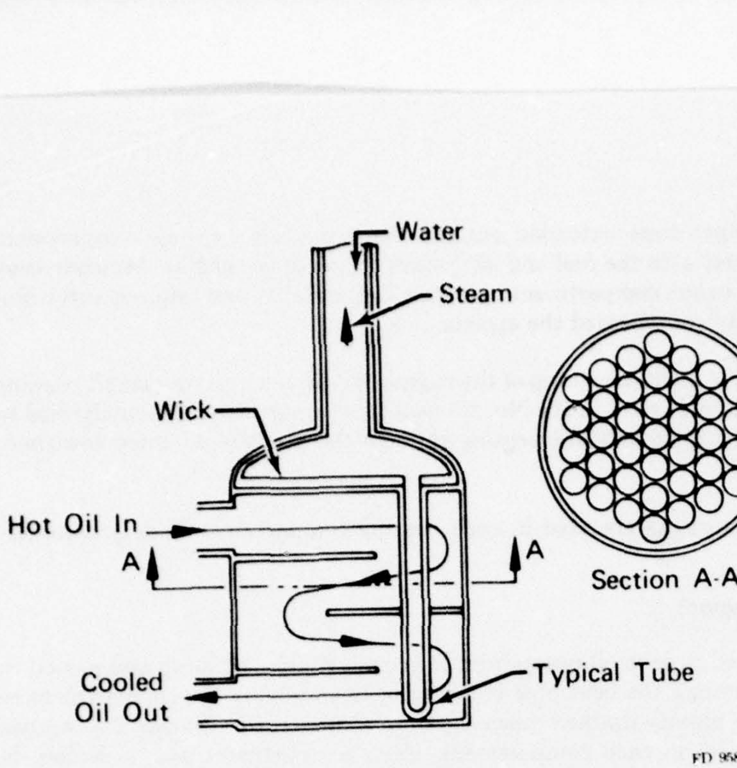

Figure 5. Heat Pipe Boiler

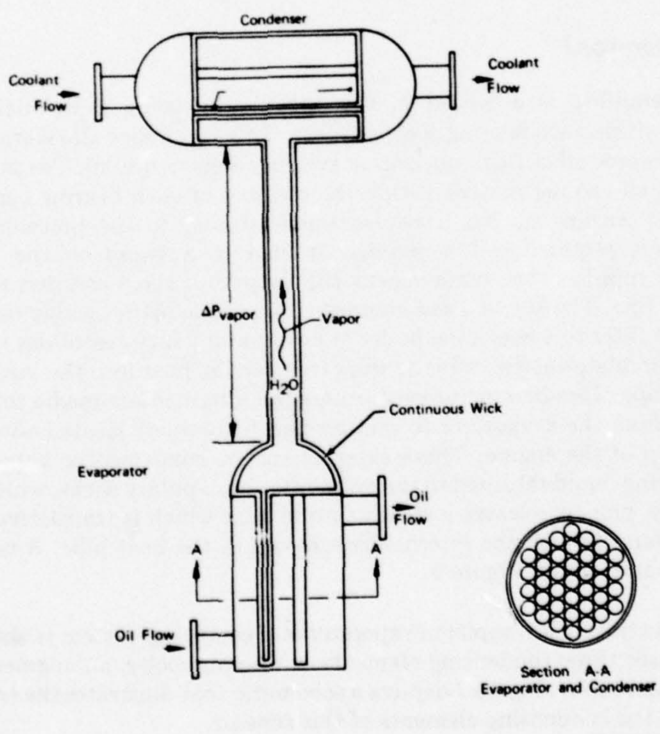

Figure 6. Evaporator and Condenser System

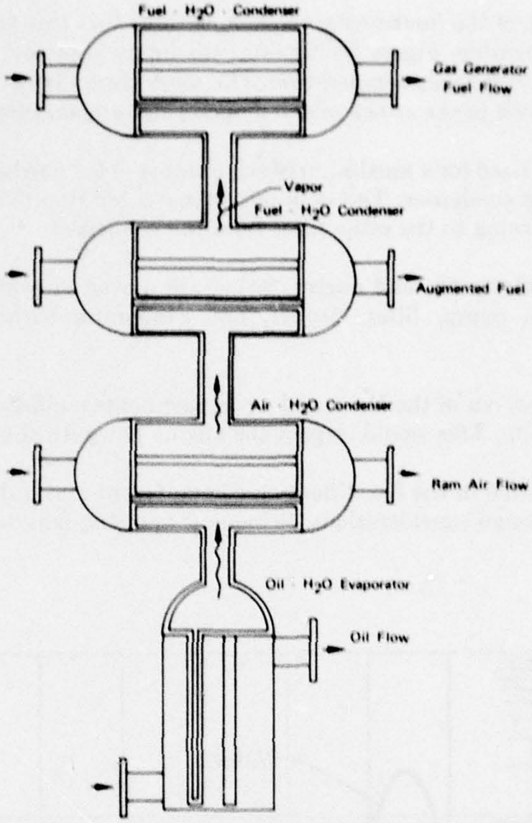

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Figure 7. Air-Fuel-H_2O Heat Pipe System

The selection of the intermediate media is influenced by the selection of media operating temperature. The intermediate media temperature must be selected between the oil temperature and the cooling fuel and air sink temperatures of the external condensers. This temperature selection trades off small evaporator size (low-media temperature) with resultant large condensers against large evaporators (high-media temperature) with resulting small condensers. The media operating temperature selected is 250°F which provides the best compromise on system design using F100-PW-100 baseline lubrication system heat generation rates.

Water was selected as the intermediate media because its heat transfer characteristics are compatible with the selected operating temperature. The total heat pipe heat transfer rate is proportional to the liquid transport factor (N_ℓ) of the intermediate media. This parameter is defined as:

$$N_\ell = (\rho_\ell \sigma h_{fg})/\mu_\ell$$

Where

ρ_ℓ = Liquid Density
h_{fg} = Latent Heat of Vaporization
σ = Evaporation Coefficient
μ_ℓ = Liquid Viscosity

The selection criteria of the intermediate media dictates that this parameter be as high as possible for system optimization. Figure 8 illustrates the liquid transport factor of various heat transfer media as a function of liquid temperature. The water media selection is based on its high liquid transport factor which peaks at the selected operating temperature.

The heat pipes were sized for a maximum Mach number of 0.3 for the water vapor traveling from the evaporator to the condenser. The wick design was sized to provide a return velocity of 1 ft/sec for the water returning to the evaporator from the condenser.

The engine burner, flowpath, and engine casing are moved outward to provide space to incorporate a gear-driven pump, filter, deoiler, and evaporator within the No. 4 bearing compartment.

Routing the heat pipes out of the No. 1 and 4 compartments requires an increase in the size of the baseline engine struts. This would impact the engine flowpath slightly.

The alternator is located in the No. 1 bearing compartment and is driven off the low rotor. Comments made under design considerations, Scheme I, apply equally to this application.

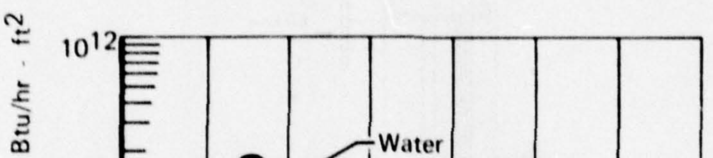


Figure 8. Liquid Transport Factor, $N\ell$, vs Temperature

The required capacity of each individual oil tank was determined by maintaining the same oil recirculation rate as the baseline engine oil tank. This was achieved for the oil tank capacities of the No. 1, 4, and 5 bearing compartments. The No. 2-3 compartment oil capacity is approximately one-half of this requirement, dictating the use of a self-contained shell structure to baffle the oil from the rotating parts or an external oil tank mounted outside the engine. A summary of the available compartment oil storage volumes, their recirculation rates, and a comparison to the baseline engine is presented below:

Scheme	*Compartment*	*Available Oil Storage Volume, gal*	*Time to Recirculate Stored Oil, sec*
III	1	0.266	8.9
	2-3	0.816	4.0
	4	0.77	8.7
	5	0.23	8.7
Baseline		3.1	8.9

*External oil tank supplies all the compartmental requirements.

d. Candidate Scheme IV

(1) Component Arrangement

This lubrication system scheme, shown in Figure 9, relocates the towershaft into the No. 4 bearing compartment to provide maximum storage volume for the oil tank in the No. 2-3 bearing compartment. The No. 4 compartment air leakage is breathed back to the gearbox through the towershaft strut. This eliminates the requirement for a boost oil pump. The alternator is located in the No. 1 bearing compartment and is driven by the low rotor. With the exception of the oil tank and alternator, all lubrication system components are located on top of the engine. The oil filter, can deaerator, deoiler, fuel/oil, and air/oil coolers are all F100-PW-100 baseline components.

A dipstick is used to determine oil level in the oil tank during servicing.

(2) System Flowpath

Oil is drawn from the oil tank through a suction line to the main oil pump mounted on top of the engine. The oil is then pumped through the oil filter to the cooling system and split to the gearbox and No. 1, 2-3, 4, and 5 bearing compartments. Oil flowpaths are provided around the pump, filter, and coolers to account for cold oil starts and a possible plugged oil filter. Scavenge pumps transfer the compartmental oil and air leakages back to the oil tank in common return lines for the No. 1, 2-3, and 5 capped compartments. These compartments eliminated the requirement for breather pipes by venting the leakage air through the scavenge pumps. The No. 4 compartment air leakage is breathed back to the gearbox through the towershaft strut while a scavenge pump is used for oil transfer back to the tank. The scavenge return is routed to the can deaerator, within the oil tank, where the air is separated from the oil. A breather line is used to transfer this air to the gearbox where it combines with the No. 4 air leakage and is vented overboard after passing through the deoiler and breather pressurizing valve.

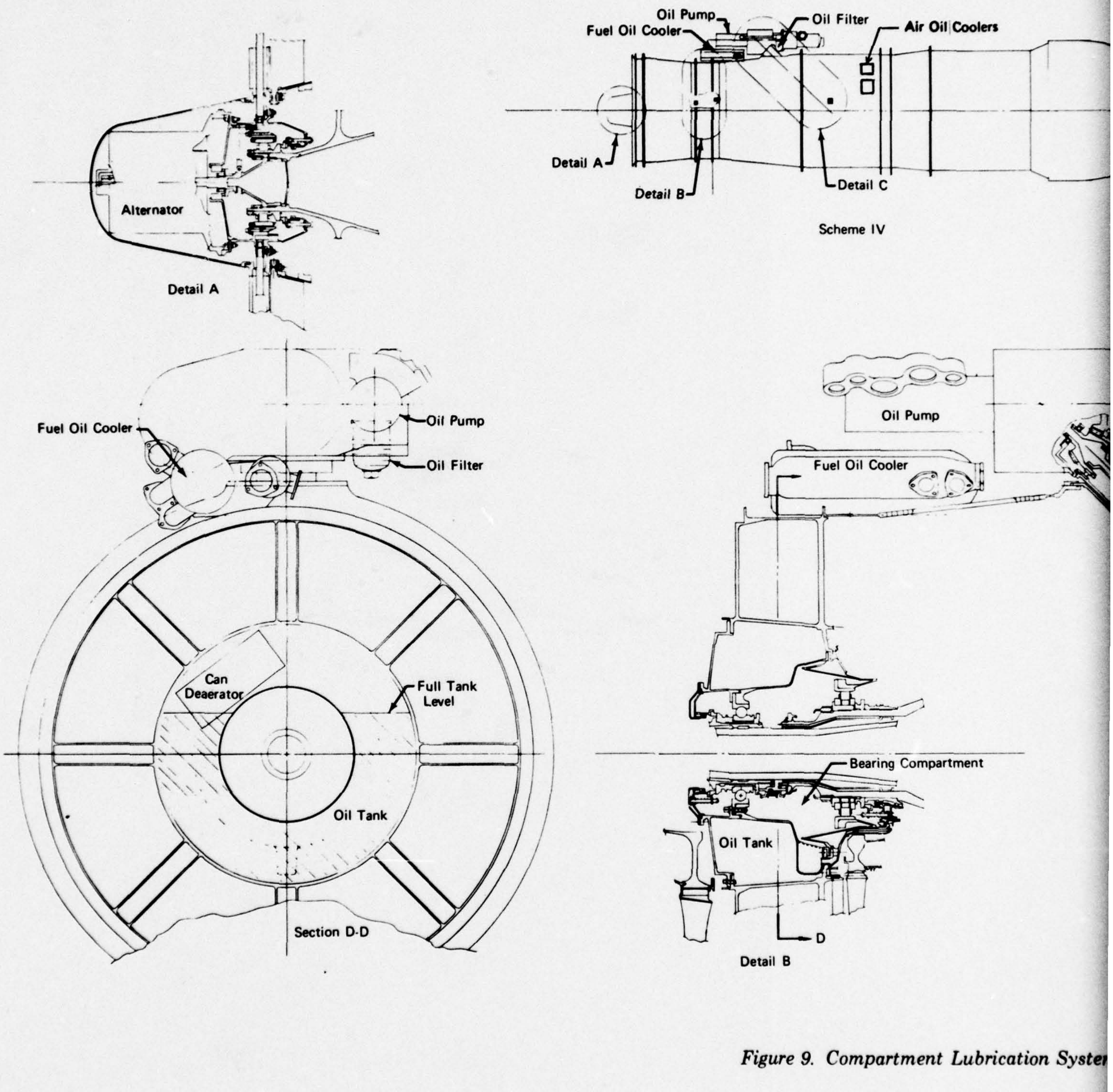
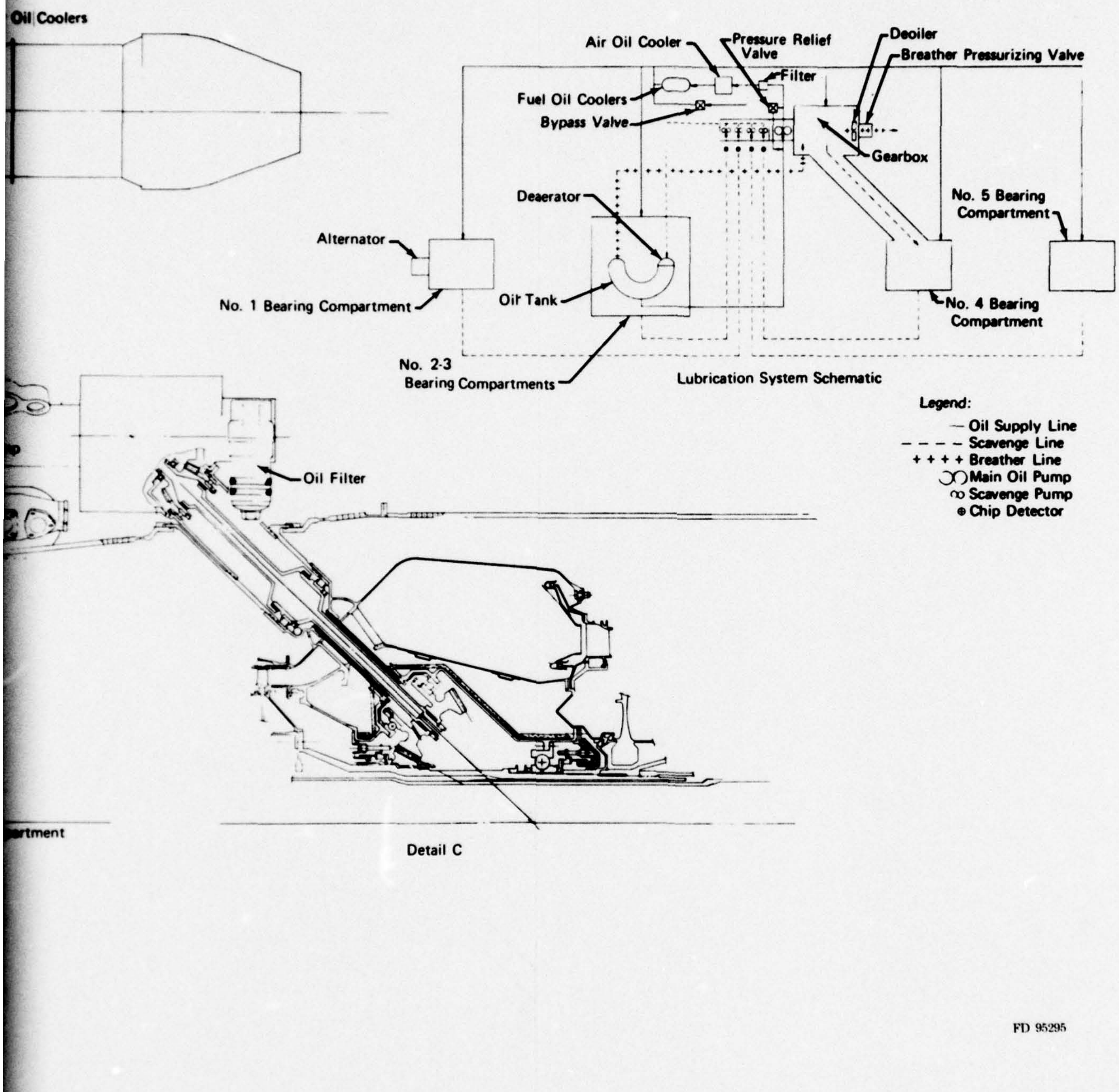


Figure 9. Compartment Lubrication Syster



t Lubrication System — Scheme IV

(3) Design Considerations

Scheme IV is consistent with I and II in that it attempts to use the No. 2-3 bearing compartment to house the oil tank to reduce vulnerability. Unlike those schemes, however, maximum space utilization is achieved in the No. 2-3 compartment by relocating the towershaft and mounting the lubrication pumps on the gearbox. The oil tank is configured using the bearing support structure to form the tank boundaries. This results in an available oil tank capacity of 3.03 gal which is sufficient to maintain required oil pressure and prevent fluctuations.

The towershaft and towershaft drive system were relocated in the No. 4 bearing compartment. This required moving the burner, gas path, and outer engine casing outward resulting in increased engine weight.

Relocating the towershaft drive into the No. 4 bearing compartment required the No. 3 mainshaft bearing to be moved along with it. The No. 3 mainshaft bearing was a ball thrust bearing which was used to maintain proper clearances between the spiral bevel gears that drive the towershaft system. Incorporating this ball bearing adjacent to the bull gear was achieved by simply switching the locations of the No. 3 and 4 mainshaft bearings. The radial gear load on the high rotor is no longer located at the front of the shaft but is now positioned mid-span. These modifications would have some minor impact on shaft critical speed characteristics.

Vulnerability is reduced by eliminating the boost pump. This is achieved by breathing the No. 4 bearing compartment back to the gearbox through the towershaft strut. The No. 4 scavenge pump size is reduced because it no longer must handle all compartment air leakage.

The alternator is located in the No. 1 bearing compartment. Comments made in Scheme I apply equally here.

e. Candidate Scheme V

(1) Component Arrangement

This lubrication system scheme locates the oil tank, main and boost oil supply pumps, and all compartment scavenge pumps in the No. 2-3 bearing compartment to reduce vulnerability as shown in Figure 10. The alternator is located in the No. 1 bearing compartment and is driven directly by the low rotor.

The gearbox is mounted on top of the engine and is driven by a towershaft off the high rotor. The towershaft is run through a support strut in the No. 2-3 bearing compartment and drives the vane lubrication pump through a gear train. A centrifugal oil filter/deoiler is mounted on the gearbox and used to filter and deaerate the oil. A breather pressurizing valve is mounted on the gearbox adjacent to the filter/deoiler to vent the compartmental air leakages overboard.

Finned wall air/oil coolers are located in the inner duct fairing. The fuel/oil coolers are plate-fin located in the fan duct wall. Chip detectors are located in the scavenge return lines. A dipstick is used for determining oil level in the oil tank during servicing.

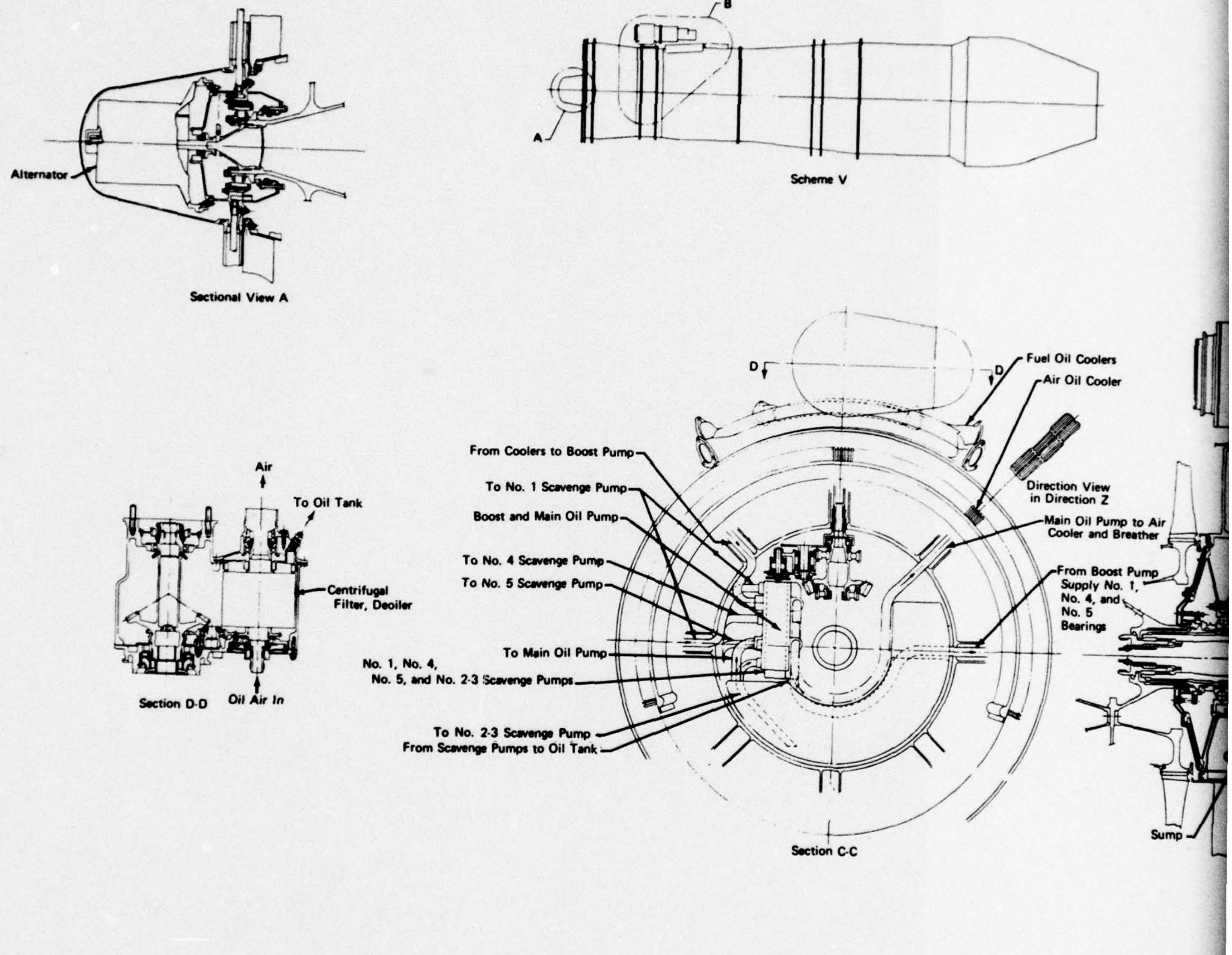
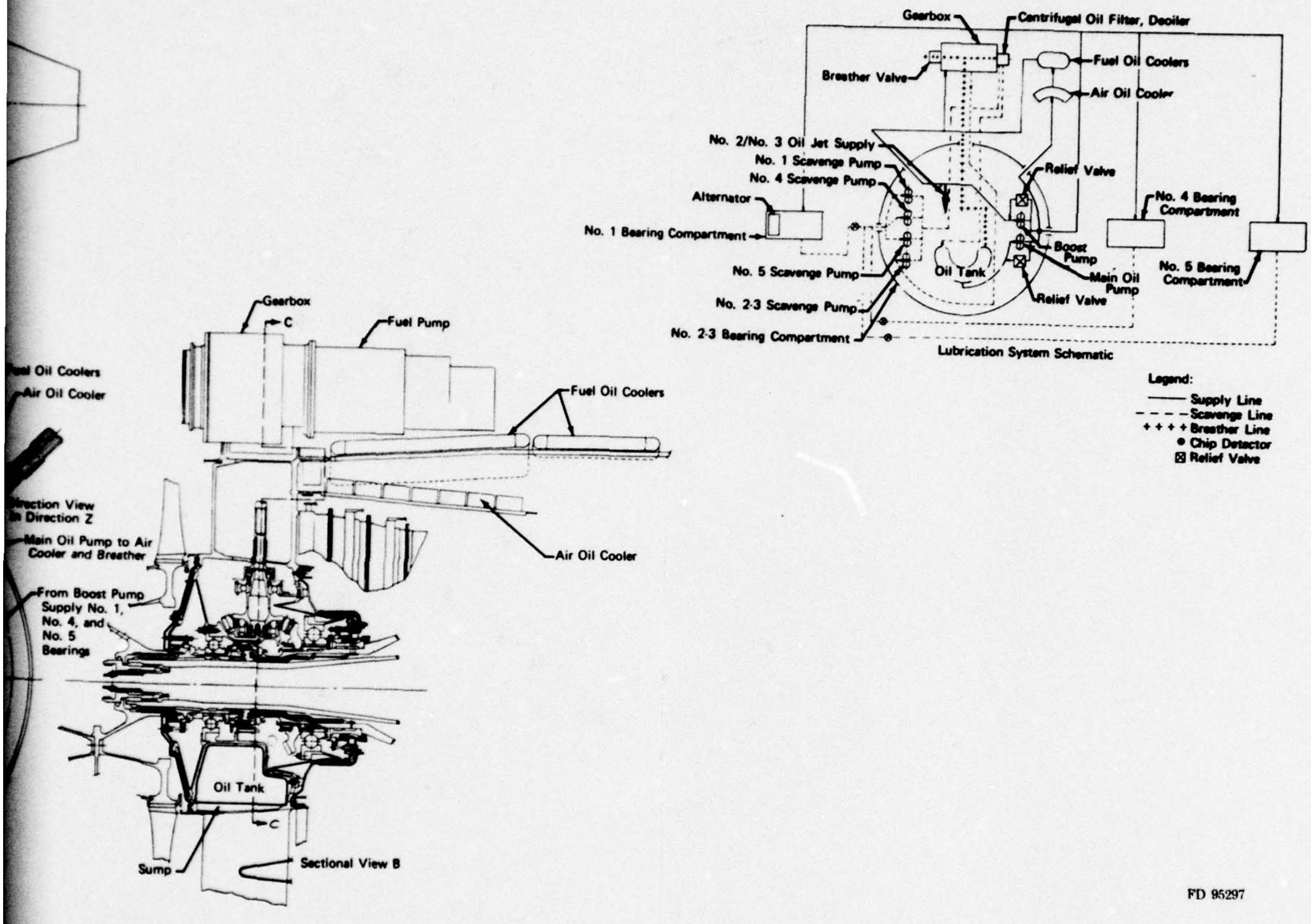


Figure 10. Compartmental Lubricat



partmental Lubrication System — Scheme V

2

(2) System Flowpath

Oil is supplied from the oil tank to the main oil pump where it is delivered to the air/oil and fuel/oil coolers outside the No. 2-3 bearing compartment. Oil bypass flowpaths are provided around the pump, filter, and coolers to account for cold oil starts and a possible plugged filter. The oil flow is then split, with one leg supplying the gearbox and the other returned back to the No. 2-3 bearing compartment. This return flow is split again, with one leg satisfying the No. 2-3 compartmental requirements and the remainder supplying the boost oil pump. From the boost pump, the oil is transferred outside the No. 2-3 bearing compartment where it is split and delivered to the No. 1, 4, and 5 compartments.

Scavenge pumps, located in the No. 2-3 bearing compartment, return the oil and air leakage from the capped No. 1, 4, and 5 compartments to the centrifugal oil filter/deoiler located on the top-mounted gearbox. One of these scavenge pumps is used to transfer oil within the No. 2-3 compartment sump to the centrifugal filter/deoiler. This oil is composed of the gearbox supply, which gravity drains down the towershaft strut, as well as the No. 2-3 compartment oil supply. The centrifugal oil filter/deoiler filters and separates oil from the air. The air is vented overboard through the breather pressurizing valve, and the deaerated oil is returned back to the oil tank located in the No. 2-3 bearing compartment.

(3) Design Considerations

In this scheme vulnerability is reduced by locating the main and boost oil pumps, all scavenge pumps, and oil tank in the No. 2-3 bearing compartment. The major mechanical difficulty with this scheme is the plumbing requirements and the available space through the compartment support struts. Vane oil pumps are used to minimize their space requirements within the No. 2-3 bearing compartment. This permits the largest oil tank capacity possible from the remaining compartment volume. Mechanical design studies indicate 1.82 gal of oil storage in this tank configuration. This is considered too small a tank capacity to meet makeup oil requirements and prevent oil pressure fluctuations with current deaeration techniques. An approach to improving the lubrication system performance with a reduced oil tank capacity is the utilization of a centrifugal filter/deaerator. Figure 11 is a schematic representation of this device. Oil is routed into the center of a hollow rotating sleeve which is dead-ended. Oil is caught into a centrifugal field, having passed through radial holes in the sleeve. Contaminants, having a higher density than oil, are centrifuged radially outward and collected in a sludge trap on the outer flow surface of the cannister. Oil collects and forms a liquid annulus which flows axially and passes over a dam restriction into an oil collection manifold from which it is routed to the oil tank. In the centrifugal field, generated by the rotating sleeve, the entrained air is forced radially inward because of its low density. Radial holes in the rotating sleeve aft of the plug section provide an escape route for the separated air which is vented overboard through a breather pressurizing valve.

The design considerations that influenced the centrifugal filter/deaerator sizing analysis are presented below:

- Total oil flow = 152 lb/min
- Scavenge oil temperature = 300°F
- MIL-L-7808 oil
- Filter rotational speed = (1.073) (high-rotor speed)

- **Contaminant composition per TDM — 2148**

  Density Carbon = 87.3 lb_m/ft^3
  ***Density of Sludge = 127.3 lb_m/ft^3**

- **Oil properties**

  Density = 53.5 lb_m/ft^3
  Viscosity = 9.5 by 10^{-4} lb/ft-sec

- **Separator bowl (rotating sleeve) is of steel material (Poisson's ratio γ = 0.3)**

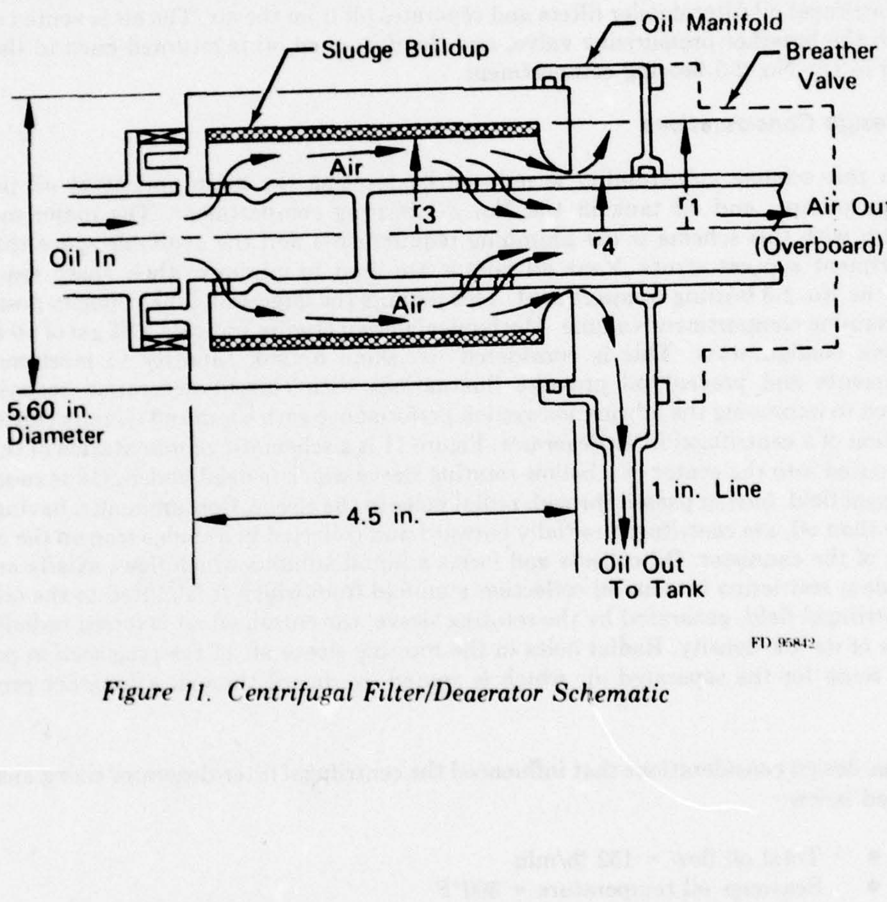


Figure 11. Centrifugal Filter/Deaerator Schematic

* Assumed to contain 90 percent organic particles with a density similar to carbon, 10 percent metal particles with a density similar to steel.

The basic geometry, which influences the micron rating, is illustrated in Figure 12 and appears in the following sizing equation:

$$L_{eff} = \left[\frac{18 \mu Q}{r_3^3 \pi \Omega^2 (\rho_p - \rho)} \right] \left[\frac{2}{(2 - (X/r_3)^2\ \delta^2} \right] \text{Constant}$$

where

L_{eff} = axial length of separation region in in. ($L_{geometric} \cong (1.1)\ L_{eff}$)

μ = oil viscosity, lb_f − sec/ft^2

Q = oil flowrate, gal/min

Ω = angular velocity of sleeve, rad/sec

ρ_p = contaminant particle density, lb_m/ft^3

ρ = oil density, lb_m/ft^3

r_3 = radius of oil ring OD, in.

X = oil ring radial thickness, in.

δ = contaminant particle diameter, microns

Constant = 1151.65×10^{10}

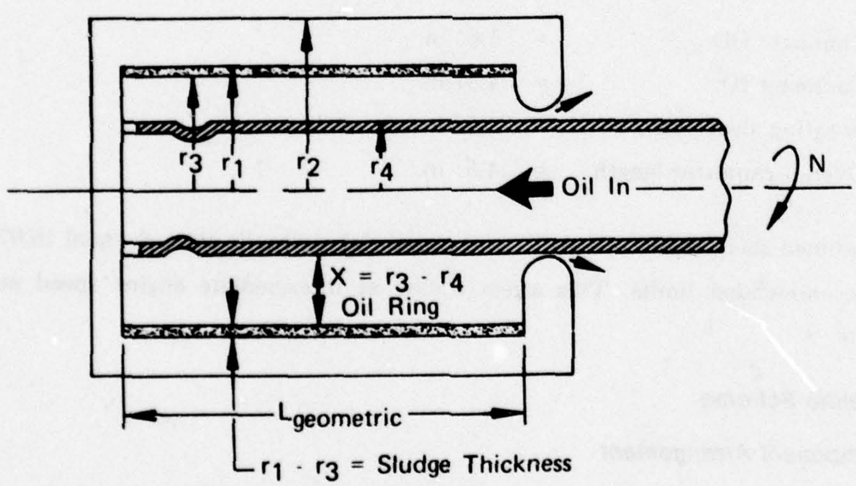


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Figure 12. Centrifugal Filter Deaerator

The time between filter cleanings can be determined by the following equation:

$$T = (\rho_{sludge})\,(V_{con})/(\dot{w}_{oil})\,(C_{onc})$$

where

T = time between cleanings, hr

ρ_{sludge} = sludge density, 127.3 lb_m/ft^3

V_{con} = filter contamination trap volume, ft^3

where $V_{con} = (L_{geo})\,(r_1^2 - r_3^2)\,\pi$

(C_{onc}) = concentration of contaminant relative to oil (lb of contaminant per lb of oil)

$\dot{w}_{oil}$ = oil flowrate, lb/hr

A summary of the centrifugal filter/deaerator predicted performance is presented below:

Filter Rating = 7.3 micron at engine idle speed (N_{sleeve} = 9871 rpm)

Time Between Cleaning = 100 hr

A summary of the selected geometry is presented below:

Cannister OD	=	5.6 in.
Cannister ID	=	4.87 in.
Rotating sleeve OD	=	2.92 in.
Overall cannister length	=	4.5 in.

Combined shell, radial hydraulic, and tangential hydraulic stresses equal 16,075 psi, well within recommended limits. This stress occurs at intermediate engine speed at sea level conditions.

f. Baseline Scheme

(1) Component Arrangement

The baseline system, illustrated in Figure 13 mounts all lubrication system components externally on the engine. Oil supply and scavenge gear pumps are gearbox mounted. The oil tank, containing the deaerator, is externally mounted above the oil pump interface. Oil filter and fuel/oil coolers are mounted externally while the air/oil coolers are mounted in the fan duct. The deoiler, engine alternator, and breather pressurizing valve are gearbox mounted.

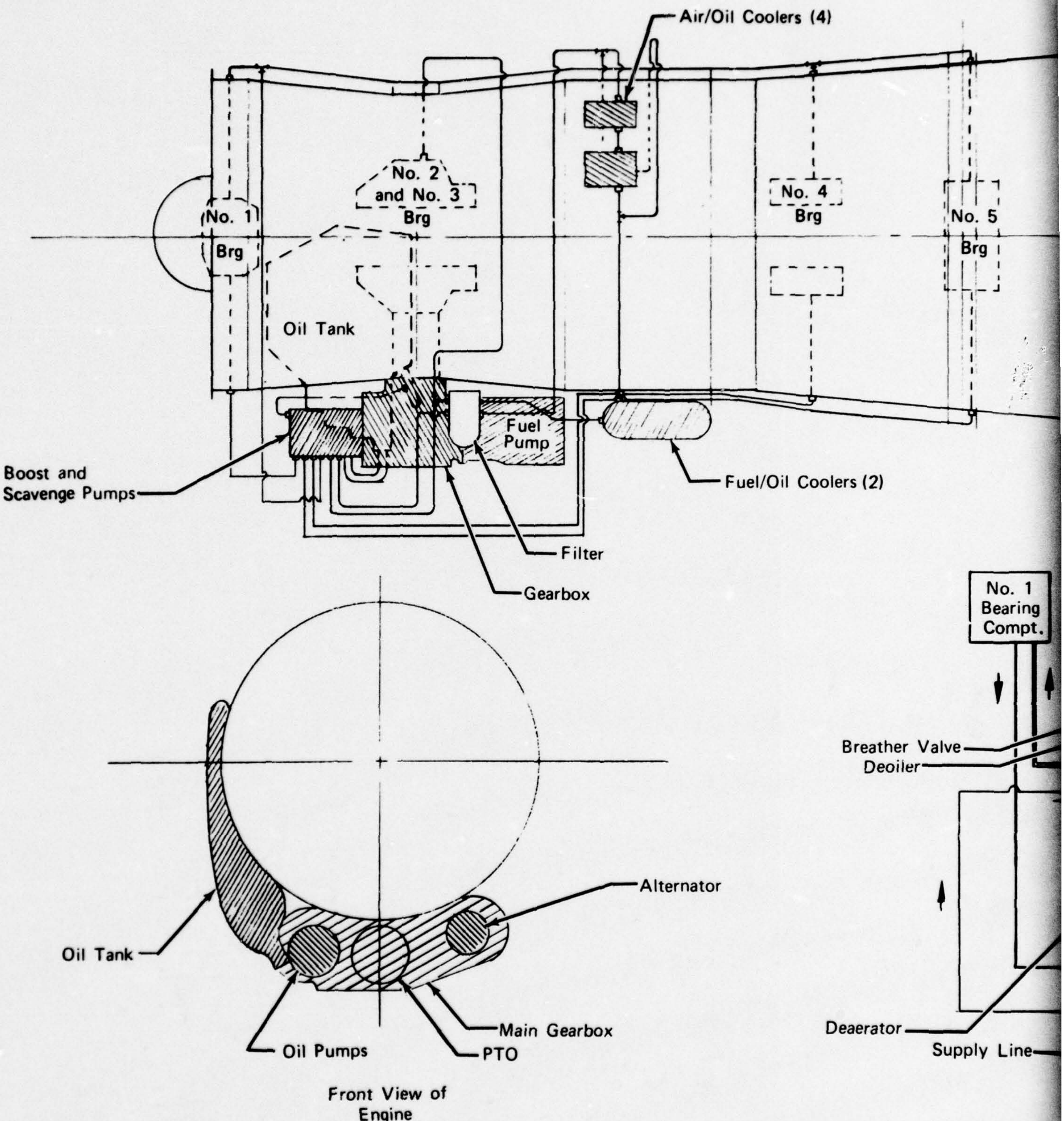
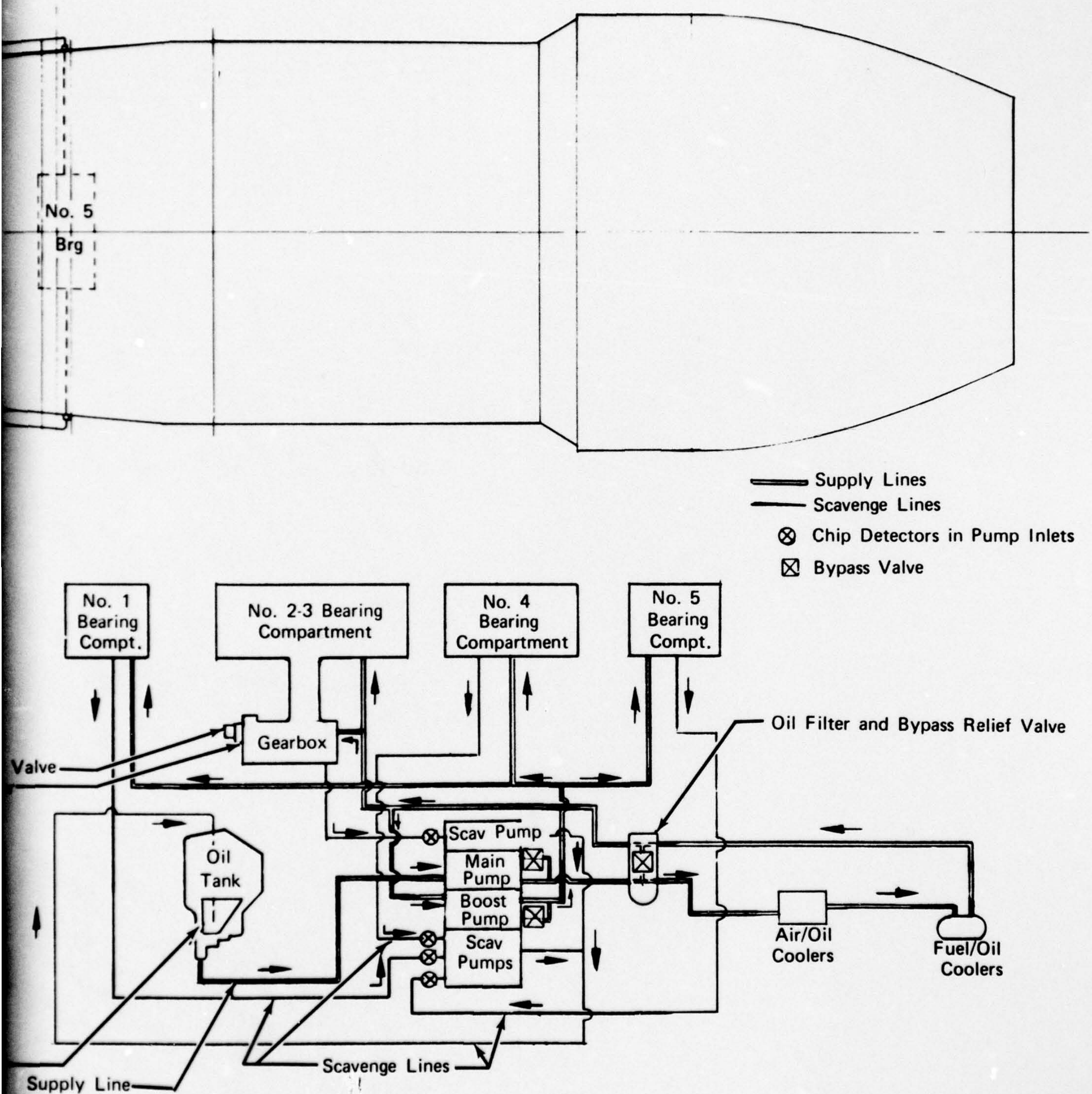


Figure 13. Baseline F100 Lu



Lubrication
System Schematic

FD 90084

Baseline F100 Lubrication System

2

(2) System Flowpath

Oil is supplied from the oil tank by a positive displacement main gear pump and passes through an oil filter before entering the oil cooling system. From the filter, oil passes through air/oil heat exchangers, then through fuel/oil heat exchangers before entering the engine. A pressure-actuated bypass valve, located in the oil filter housing, bypasses oil around all heat exchangers during cold oil start conditions to prevent excessive heat exchanger pressure drop. A portion of this oil then goes to the No. 2-3 compartment and gearbox. A second positive displacement gear pump, located downstream of the heat exchangers, pumps the remaining portion of the oil to the No. 1, 4, and 5 bearing compartments. This pressure boost is required because of the potentially higher compartment pressures associated with the scavenge oil breather system. The scavenge system for these compartments does not use separate breather lines for venting air leakages. Instead, it scavenges both the air and oil together in a single line to the scavenge pump. Oil flow is accurately distributed to each desired location within the bearing compartments and gearbox by metering jets.

Oil is returned from the engine compartments by gear scavenge pumps externally located on the gearbox with the boost and main pumps. A breather valve, located in the gearbox, regulates gearbox, No. 2-3 bearing compartment, and oil tank pressure by venting breather airflow to ambient.

3. Quantitative Evaluation of Candidate Lubrication System Schemes Leading to Phase I System Selection

a. General

Table 4 presents the results of the quantitative evaluation of the five compartmental lubrication system schemes on the basis of vulnerability, maintainability, reliability, acquisition costs, life cycle costs, weight, frontal area, manufacturing, assembly, and development considerations, and system compromises. The point weighting of the criteria (Table 2) and the method of analysis were presented previously. All analyses were performed on a differential basis, compared with the baseline F100-PW-100 engine. Also, where practical, the analyses were performed on a component basis so that all of the schemes could be reviewed to ascertain which components from other schemes could be used to further improve the winning scheme. A discussion of the results of each analysis follows.

TABLE 4. SUMMARY RATING

Rating Criteria	*Vulnerability*	*Maintainability*	*Reliability*	*Acquisition Costs*	*Life Cycle Costs*	*Weight*	*Frontal Area*	*Manufacturing, Assembly, and Development Difficulties*	*System Compromises*
Maximum Point Allotment	*30*	*25*	*10*	*5*	*5*	*10*	*8*	*3*	*4*
Scheme									
I	25.2	18.2	2.9	4.8	4.8	9.6	8.0	1.0	0.4
II	22.9	24.4	10.0	4.5	4.7	8.5	8.0	0.8	0.4
III	30.0	8.9	0	2.3	2.6	6.9	7.7	0.3	0.3
IV	20.6	0	3.5	4.7	4.5	8.2	7.7	0.8	0.7
V	21.6	10.6	2.0	3.8	4.1	8.4	8.0	0.4	0.4
Baseline	19.0	25.0	3.1	5.0	5.0	10.0	7.5	3.0	4.0

b. Vulnerability

A comparison of the Δ vulnerable areas to the baseline engine and supporting numbers for the vulnerability ratings of Table 4 are presented in Appendix B.

The presentation of the vulnerable area calculations for each lubrication system component for the five schemes is too lengthy for this report. However, a brief summary of the reasons governing the point allotment of each is as follows:

1. *Scheme I* —"A" kills are reduced by placing fuel/oil coolers on top of the engine. A kill vulnerability within the bearing compartments is much the same as the other schemes (except IV).

   "B" kills are reduced from the baseline by placing the oil tank, pumps, and filter in the No. 2-3 bearing compartment and positioning the gearbox on top of the engine.

2. *Scheme II* — A kills are increased significantly due to the much larger projected area of the plate-fin fuel/oil cooler which was proposed.

   "B" kills are greatly increased due to the finned wall air/oil coolers having a larger projected area. The fan ducts do not provide a significant amount of protection for these coolers.

   As a result of the blowdown scavenge systems, the projected areas of the No. 1, 4, and 5 bearing compartments (top, bottom, and side views only) are smaller, which provide for lower vulnerability. The elimination of the No. 1, 4, and 5 scavenge pumps also contributes to this.

3. *Scheme III* — "A" kill vulnerability is reduced by placing the (fuel) heat exchangers on top of the engine.

   The heat pipe scheme resulted in all of the oil and oil system components being contained in individual bearing compartments. This, in itself, reduces the "B" kill probability. Also, it is estimated that only 20 percent of the baseline external plumbing is required, which provides for considerably less vulnerable area. (A hit to a heat pipe was not termed critical enough to constitute a "B" kill.)

4. *Scheme IV* — The "A" kill vulnerability is just slightly less than baseline only because of the gearbox being on top of the engine.

   "B" kills are higher than other schemes because of the external oil pumps and filter, although, the fuel/oil cooler on top of the engine reduces this somewhat.

5. *Scheme V* — "A" kill vulnerability is very similar to Scheme II (with plate-fin fuel/oil cooler).

   The "B" kill vulnerability is increased over other schemes due to the external oil filter which requires increasing the gearbox size. Again, placing the oil tank and pumps in the No. 2-3 bearing compartment greatly reduces the vulnerability from baseline.

c. Maintainability

The results of the maintainability analysis, detailed to the component basis, are presented in Appendix C. A summary of the differential systems maintenance man-hours per million engine flight hours (Δ MMH/10^6 EFH), compared to the baseline F100-PW-100 engine, is shown below:

Scheme	MMH/10^6 EFH Over Baseline
I	25,485
II	2,475
III	60,773
IV	94,253
V	54,202

A brief summary of the maintainability features and penalties of each scheme is as follows:

1. *Scheme I* — The location of the main oil and scavenge pumps results in an increase in MMH/EFH from baseline. This is due to additional task times and a higher parts discrepancy rate with the main oil pump inside the No. 2-3 bearing compartment, and scavenge pumps in the individual No. 1, 4, and 5 compartments.

2. *Scheme II* — The blowdown scavenge system used for the No. 1, 4, and 5 bearing compartments significantly reduces the MMH/EFH in this scheme. Without the need of carbon seal assemblies and their supports, the total task times for those compartments are greatly reduced. The elimination of the No. 1, 4, and 5 scavenge pumps is also a major contributing factor.

   The items that increase the MMH/EFH most, due to additional task times required, are the main oil pump, No. 2-3 scavenge pump, and the finned wall air/oil cooler. There is also a higher parts discrepancy rate for the pumps being inside the No. 2-3 bearing compartment.

3. *Scheme III* — With pumps, filters, and deoilers to remove/replace in each bearing compartment, the task times required to maintain these components are increased greatly over the baseline.

   The frequency of part discrepancies for individual pumps, filters, etc., is higher than the frequency for the single part in the baseline, which performs the same job.

4. *Scheme IV* — The total Δ MMH/EFH is higher in this scheme, mostly due to the increase in task times required for the high compressor rotor and stator assembly, No. 4 bearing compartment, and diffuser case. There is a significant decrease in task times for the No. 2-3 bearing compartment due to the shifting of the gearbox drive mechanism to the No. 4 compartment. Many of the parts discrepancy rates in this scheme are the same as for baseline since the oil pumps, filter, and coolers are on the outside of the engine.

5. *Scheme V* — The finned wall air/oil coolers increase task times significantly, since the fan ducts have to be removed to obtain access to them. The pumps in the No. 2-3 bearing compartment also increase the task times, as well as the parts discrepancy rates.

d. Reliability

Reliability calculations, detailed to the component basis, are presented in Appendix C along with the maintainability figures. A summary of the differential system part discrepancies per million engine flight hours (Δ discrepancies/10^6 EFH) compared to the baseline F100-PW-100 engine, is as follows:

Scheme	*Δ Discrepancies/10^6 EFH Compared to Baseline*
I	+44
II	−1476
III	+670
IV	−86
V	+252

Note that Scheme II has the highest reliability rating, and Scheme III has the worst rating. A summary of the reliability features and penalties of each scheme is as follows:

1. *Scheme I* — Incorporation of the oil pumps into the bearing compartments resulted in a small decrease in reliability due to the increased number of parts involved, but this was partially offset by using the No. 2-3 compartment as an oil reservoir. The net effect was a small decrease in reliability.

2. *Scheme II* — The greatest improvement in reliability was calculated for the blowdown scavenge system. The major contributing factors are the elimination of the bearing compartment carbon face seals and the No. 1, 4, and 5 compartment scavenge pumps.

   Incorporating the pumps into the No. 2-3 compartment reduced reliability by a slight amount due to the increased number of parts in the drive system.

3. *Scheme III* — The increased complexity of this scheme, caused by the incorporation of heat pipe evaporators and condensers, resulted in Scheme III having the lowest reliability of the five schemes.

4. *Scheme IV* — There were no significant differences in reliability from the baseline engine. Minor improvements can be attributed to the incorporation of the oil tanks into the compartment and the reduced complexity of the No. 2-3 compartment. The reliability of the accessory drive system was reduced by a small amount due to the added complexity and additional parts required to drive the accessory section from the No. 4 compartment. The net effect was a slight improvement in reliability.

5. *Scheme V* — As noted in Scheme I, incorporation of the pumps into the bearing compartment causes a decrease in reliability. A relatively large reduction was caused by the addition of an oil boost pump, which was not employed in the other schemes.

e. Acquisition Costs

The baseline F100-PW-100 engine was found to have the lowest lubrication system total acquisition cost and was awarded the maximum point allotment of five. A breakdown of the component costs for each scheme is presented in Appendix D. The total increase in cost for each scheme over the baseline engine is as follows:

Scheme	*Δ Cost Over Baseline*
I	+$ 943
II	+$ 3,469
III	+$36,907
IV	+$ 1,679
V	+$ 9,183

f. Life Cycle Costs

The following summary shows that the life cycle costs for all five schemes exceeded that of the baseline F100-PW-100 engine.

Scheme	*Δ Cost From Baseline $ (Millions)*
I	+ 3.7
II	+ 4.7
III	+66.0
IV	+ 7.0
V	+16.4

Visibility into the generation of life cycle cost values and calculation of the rating points can be obtained from the detailed values presented in Appendix E.

g. Weight

All five candidate schemes were found to weigh more than the baseline (F100) lubrication system. A summary of the increase in weight over the baseline engine is given below:

Scheme	*Δ Weight From Baseline (F100) (lb)*
I	+ 15
II	+ 61
III	+150
IV	+ 75
V	+ 63

A detailed breakdown of component weights for each scheme is given in Appendix F.

h. Frontal Area

The overall variation in frontal area was very small for the five schemes. However, as shown below, the projected frontal area for all five compartmental lubrication schemes is slightly less than that of the baseline F100-PW-100 engine.

Scheme	Δ Frontal Area From Baseline in.^{2}
I	−99.6
II	−99.6
III	−45.0
IV	−42.2
V	−99.6

I. Manufacturing, Assembly, and Development Considerations

This criterion was evaluated by first listing the manufacturing, assembly, and development difficulties associated with each scheme. Each difficulty was then assigned a numerical value of −1 to −10, based on the severity of the problem, with the worst problems receiving a −10. Appendix G provides a tabulation of these difficulties and the points for each. A summary of the total points assessed against each scheme is shown below.

Scheme	Total Points Assessed Against Scheme
I	− 9
II	−12
III	−27
IV	−12
V	−21
Baseline	− 3

The scheme with the minimum negative points was assigned a comparison to best scheme factor of (1) and was given the maximum rating points (3) assigned to this criterion. All other schemes received fewer rating points proportionally to the number of negative points assessed against them.

J. System Compromises

A list of lubrication system compromises associated with each candidate scheme and the baseline engine is given in Appendix G. The severity of each compromise was rated from −1 to −10, with the most severe problem receiving a rating of −10. A summary of the total points assessed against each scheme is as follows:

Scheme	Points
I	−38
II	−41
III	−63
IV	−24
V	−38
Baseline	− 4

The maximum of four points for this criterion was assigned to the baseline scheme since it had the least number of negative points against it. All other schemes received a proportionate value of these points, based on a numerical ratio of the absolute value of total points compared to the best (baseline) scheme.

k. Results of Phase I Trade Studies

Scheme II was determined to be the best of the advanced lubrication systems, as well as superior to the F100-PW-100 baseline system as evaluated in Phase I of this program. Table 5 lists the various study schemes, along with their final point totals. A detail breakdown of the individual scores in each rating criterion for each scheme was presented in the preceding section.

TABLE 5. OVERALL POINT RATING SUMMARY

Scheme	*Point Totals*
I	74.8
II*	84.2
III	53.0
IV	50.7
V	59.3
Baseline	81.6

* Scheme II as defined in Section II.2.b.

A review of Scheme II on a component basis indicated that improvement in its competitive position could be achieved by substituting baseline F100-PW-100 oil coolers for the finned wall air/oil and plate-fin fuel/oil coolers. This modification (Scheme II-1) has the following impact, relative to the original Scheme II system, on the criteria summarized below:

1. *Maintainability* — Scheme II-1 reduces maintainability by 9,798 maintenance man-hours per million engine flight hours over Scheme II.

2. *Vulnerability* — Scheme II-1 is 16.6 percent less vulnerable than Scheme II.

3. *Cost* — Scheme II-1 reduces life cycle costs $6.3 million and acquisition costs $6,761 per engine when compared to Scheme II.

4. *Weight* — Scheme II-1 is 58.6 lb lighter than Scheme II.

The basic Scheme II system was found superior to the baseline F100-PW-100 system in vulnerability, reliability, and frontal area, as shown in Table 4 in the previous section. Scheme II-1 further improved its competitive position against the F100-PW-100 system by being determined superior to the baseline system in the maintainability, acquisition cost, and life cycle cost criteria categories.

The selection of Scheme II provided several areas of technology that will be useful in future engine applications. These are:

- High-speed oil supply and scavenge pumps running two and one-half times conventional engine pump speeds.

- High-speed, compact drive gear train.

- Oil deaeration improvements in conjunction with a small volume oil tank.

- Investigation of assembly and servicing techniques required in an advanced engine which utilizes a compartmental lubrication system.
- Provided for the evaluation of blowdown scavenge system analysis for military aircraft.
- Provided for evaluation of oil handling characteristics in compact bearing compartment applications.

SECTION III
PHASE II — DETAILED EVALUATION AND PRELIMINARY DESIGN OF SELECTED SYSTEM

1. PHASE I — REVIEW AND UPDATING OF INITIAL QUANTITATIVE ANALYSIS

A review of the Compartmental Lubrication System program was held at Pratt and Whitney Aircraft, Government Products Division on March 22, 1976 through March 25, 1976 with the AFAPL Project Engineer. It was decided during this review that the method for calculating criteria rating points for reliability and maintainability in the Phase I analysis was not consistent with the method used for the other rating criteria. A reevaluation of the points (Table 6) showed that Scheme II was still the top candidate compartmental lubrication system scheme but it no longer rated higher than the baseline F100-PW-100 system as was reported in the initial Phase I results.

The selected compartmental lubrication system scheme (Scheme II) was then revised based on knowledge obtained from the Phase I quantitative analyses. Since these analyses were done on a component basis, it was possible to select components from other schemes to improve the selected scheme. These modifications and design refinements included replacing the fin-wall air-oil heat exchanger with fan duct plate-fin modules, replacing the plate-fin fuel-oil heat exchangers with shell and tube heat exchangers and moving the oil filter external on top of the engine to provide more oil tank volume in the No. 2-3 compartment. These revisions were incorporated in the preliminary design layout (Phase II, Task I) shown on Figure 14. The quantitative analysis was then repeated incorporating these revisions, and Scheme II was found to be significantly better than the baseline system or any of the other candidate schemes as shown in Table 7.

TABLE 6.
QUANTITATIVE TRADE STUDIES

Rating Criteria	*Vulnerability*	*Maintainability*	*Reliability*	*Acquisition Costs*	*Life-Cycle Costs*	*Weight*	*Frontal Area*	*Manufacturing Assembly and Development Difficulties*	*System Compromises*	*Totals*
Maximum Point Allotment	*30*	*25*	*10*	*5*	*5*	*10*	*8*	*3*	*4*	*100*
Scheme										
I	25.2	11.5	8.2	4.8	4.8	9.6	8.0	1.0	0.4	73.5
II	22.9	22.5	10.0	4.5	4.7	8.5	8.0	0.8	0.4	82.3
III	30.0	6.6	7.7	2.3	2.6	6.9	7.7	0.3	0.3	64.4
IV	20.6	4.7	8.4	4.7	4.5	8.2	7.7	0.8	0.7	60.3
V	21.6	7.2	8.0	3.8	4.1	8.4	8.0	0.4	0.4	61.9
Baseline	19.0	25.0	8.3	5.0	5.0	10.0	7.5	3.0	4.0	86.8

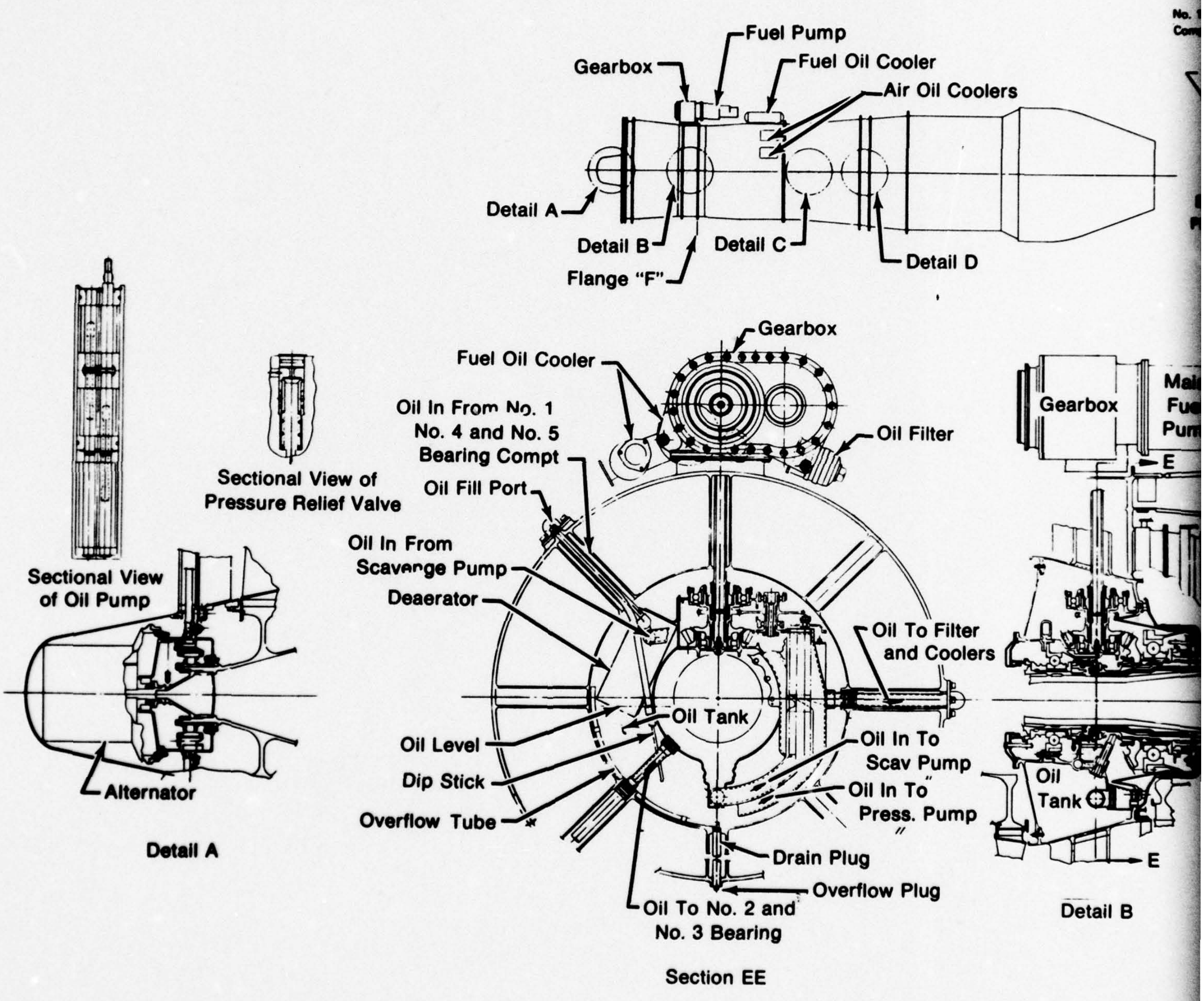
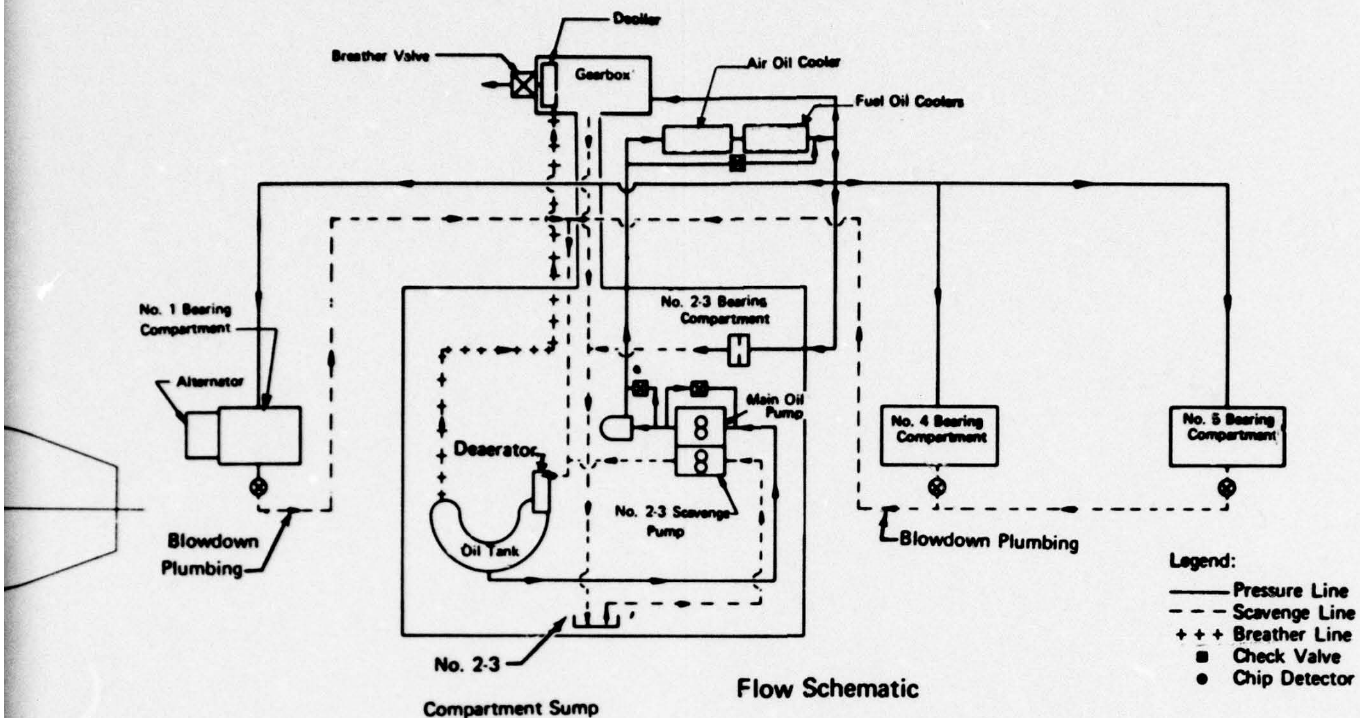
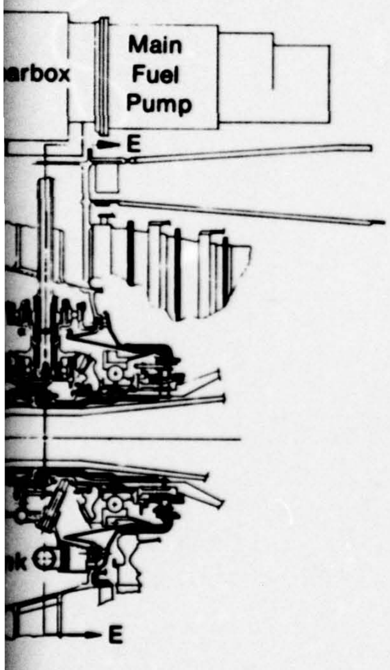


Figure 14. Preliminary Des



Flow Schematic



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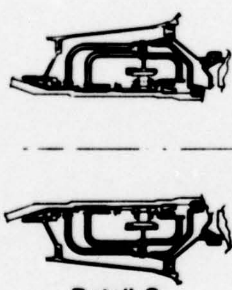

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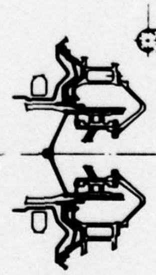

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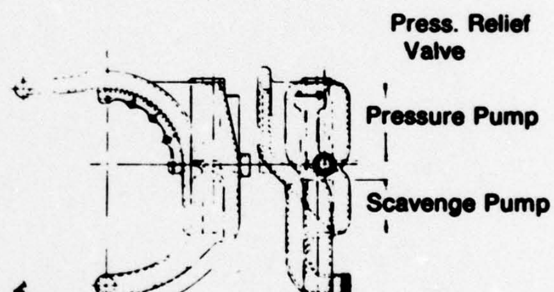


View of Pump Housing

eliminary Design Layout

TABLE 7
QUANTITATIVE EVALUATION WITH REVISIONS
IN SCHEME II

Rating Criteria	Vulnerability	Maintainability	Reliability	Acquisition Costs	Life-Cycle Costs	Weight	Frontal Area	Manufacturing Assembly and Development Difficulties	System Compromises	Totals
Maximum Point Allotment	30	25	10	5	5	10	8	3	4	100
Scheme										
I	25.2	6.9	8.5	4.3	4.4	9.6	8.0	1.0	0.4	68.3
II	27.4	25.0	10.0	5.0	5.0	9.9	8.0	1.8	0.5	92.6
III	30.0	3.9	7.9	2.0	2.3	6.9	7.7	0.3	0.3	61.3
IV	20.6	2.8	8.6	4.2	4.1	8.2	7.7	0.8	0.7	57.7
V	21.6	4.3	8.3	3.4	3.7	8.4	8.0	0.4	0.4	58.5
Baseline	19.0	14.9	8.5	4.5	4.5	10.0	7.5	3.0	4.0	75.9

2. FINAL SELECTION OF COMPARTMENTAL LUBRICATION SYSTEM CONFIGURATION

a. Design Considerations

The intent of the selected compartmental lubrication system is to provide reduced vulnerability by using the major bearing compartment to house and shield the critical lubrication system components. The oil tank, being the largest and most vulnerable component, was the prime candidate for inclusion in the No. 2-3 bearing compartment. In order to also include the oil supply pump and No. 2-3 scavenge pump in this compartment without overly restricting oil tank volume, it was necessary to increase pump speed to 10,000 rpm. This is two and one-half times the speed of conventional gas turbine engine oil pumps. The increased speed provides for a smaller pump volume and, in addition, allows a smaller gear set for speed reduction from the 26,000 rpm towershaft drive to the pump. The 2.8 gal oil tank volume was initially considered marginal, but rig tests were run and substantiated deaeration capabilities.

The gearbox is mounted on top of the engine and driven by a towershaft running through a vertical support strut in the No. 2-3 compartment. Running the towershaft through the top of the engine provides more room in the bottom of the compartment for the oil tank. Mounting the gearbox on top of the engine also reduces vulnerability to ground fire and missile shrapnel. The deoiler and breather pressurizing valve as well as the alternator are gearbox-mounted. The alternator was located in the No. 1 compartment bullet nose during the Phase I analysis to satisfy a statement-of-work requirement for an internal location for the alternator drive. However, the alternator was moved back to the gearbox due to problems with supplying power to the engine during starting with the low-rotor-driven alternator. Quantitative analysis has also shown that the bullet nose location did not offer any improvement in vulnerability and resulted in a slight increase in cost and weight. Shell and tube fuel-oil coolers and a nonbypassing oil filter are also located on top of the engine to reduce vulnerable area. Plate-fin, air-oil coolers are mounted in the fan duct at the top of the engine.

b. Scavenge Options for No. 1, 4, and 5 Bearing Compartments

One major concern with the selected compartmental lubrication system scheme was the lack of substantiation for the blowdown system used to scavenge the No. 1, 4, and 5 bearing compartments. This type scavenge system has been used successfully on subsonic engines such as the JT15D but has not been attempted on engines for supersonic aircraft. An analytical

simulation of the scavenge blowdown system (see Appendix H) shows that the system can be sized to function without pressure reversals (oil leakage) during transient decels utilizing either labyrinth or carbon seals in the compartments. However, labyrinth seal leakage would be approximately 1000 lb per hour which is out of the range of oil tank deaeration capabilities and would result in an unnecessary performance penalty on the engine. Consequently, it was decided to reevaluate the selected compartmental lubrication system quantitatively with four (4) optional methods of scavenging the No. 1, 4, and 5 compartments. The Phase I evaluation criteria was again used to maintain a constant base for the quantitative analyses. Optional scavenging methods are as follows:

Option I	-	The No. 1, 4, and 5 compartments are scavenged by a blowdown system, but carbon seals are used in these compartments to limit air leakage.
Option II	-	The No. 1, 4, and 5 compartments are scavenged by gear pumps on the top mounted gearbox. Carbon seals are used in these compartments.
Option III	-	The No. 1 and 5 compartments are scavenged by gear pumps mounted within their respective compartments, and the No. 4 compartment is scavenged by a gear pump mounted within the gearbox. All compartments incorporate carbon seals.
Option IV	-	The No. 1, 4, and 5 compartments are scavenged by gear pumps on the top mounted gearbox like Option II. However, labyrinth seals are used for the No. 1, 4, and 5 compartments, and the volumetric displacement of the scavenge pumps is used to limit seal leakage. Labyrinth seals were not used in the No. 2-3 compartment since seal leakage could not be limited by this compartment's scavenge pump. The No. 2-3 compartment is breathed to ambient through the gearbox breather valve which could result in sufficient air leakage by the labyrinth seals to cause foaming in the gearbox and/or the bearing compartment.

c. Comparison of Configuration Options Using Phase I Criteria

Table 8 shows that the Option I, II, and III total point allotments are all within 5.3 points of the baseline F100-PW-100 engine. However, Option IV with 93.4 points is 11.9 points (15%) better than the baseline engine. This is primarily due to the improved maintainability of the labyrinth seals over carbon seals and the reduction in vulnerability with the oil tank inside the No. 2-3 compartment.

Based on this analysis the scavenge system utilizing high-speed gear pumps and labyrinth seals for the No. 1, 4, and 5 compartments was incorporated in the final compartmental lubrication system design. The final design layout is shown in Figure 15, and the evaluation of this system compared with the baseline F100-PW-100 engine is presented in the following paragraphs.

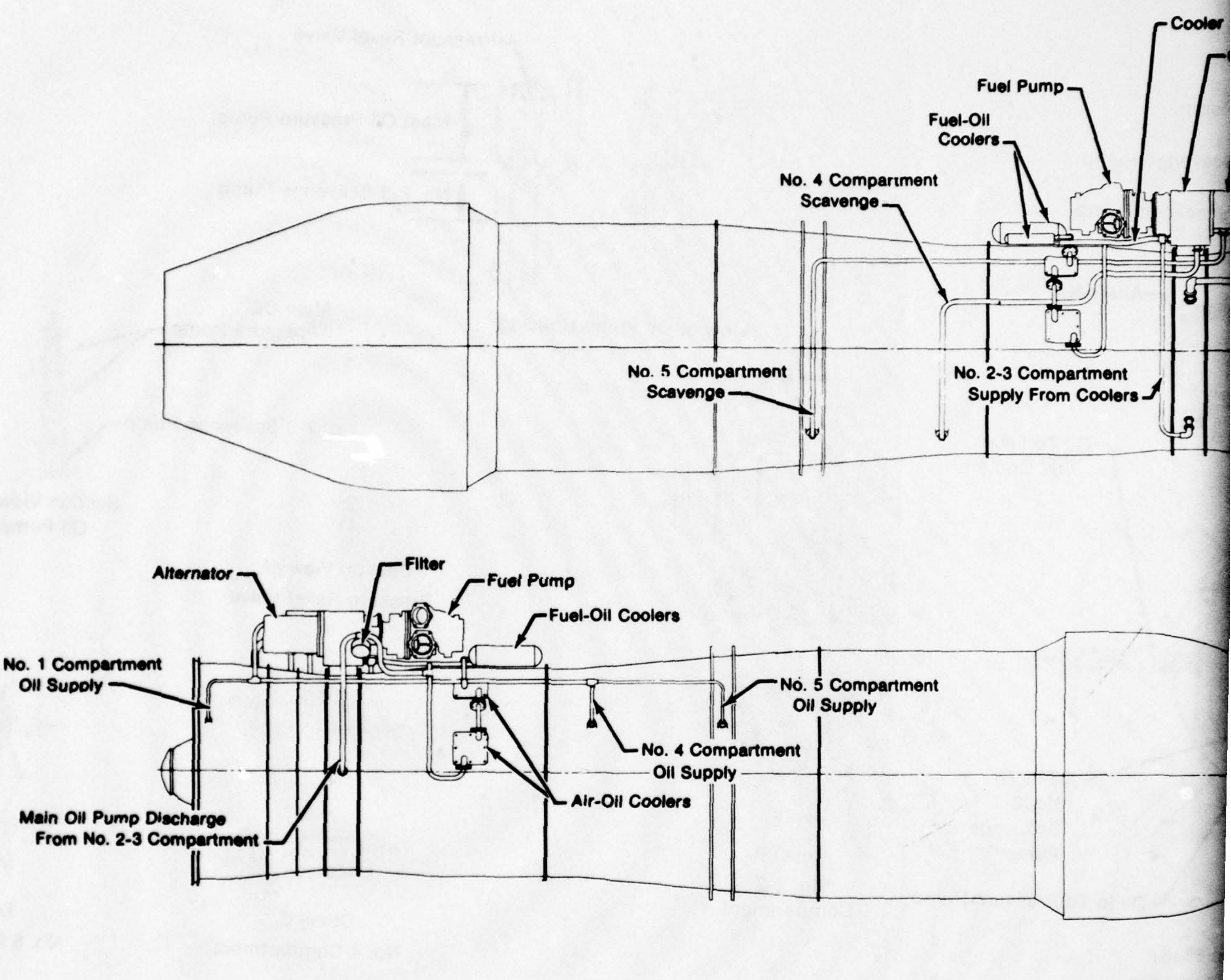
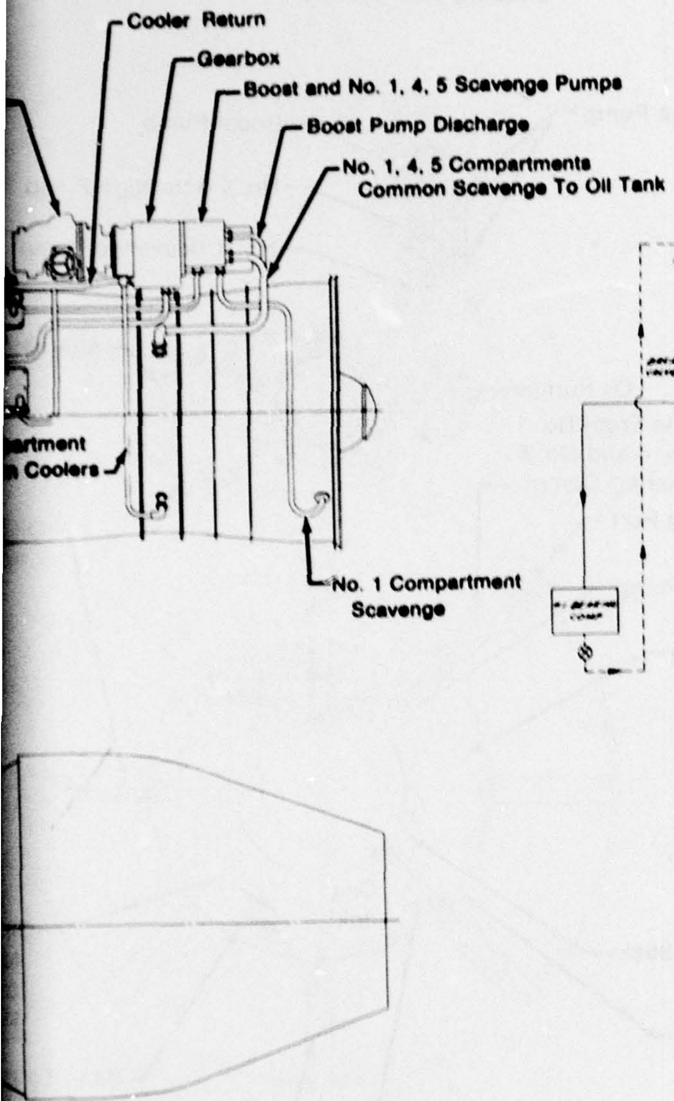
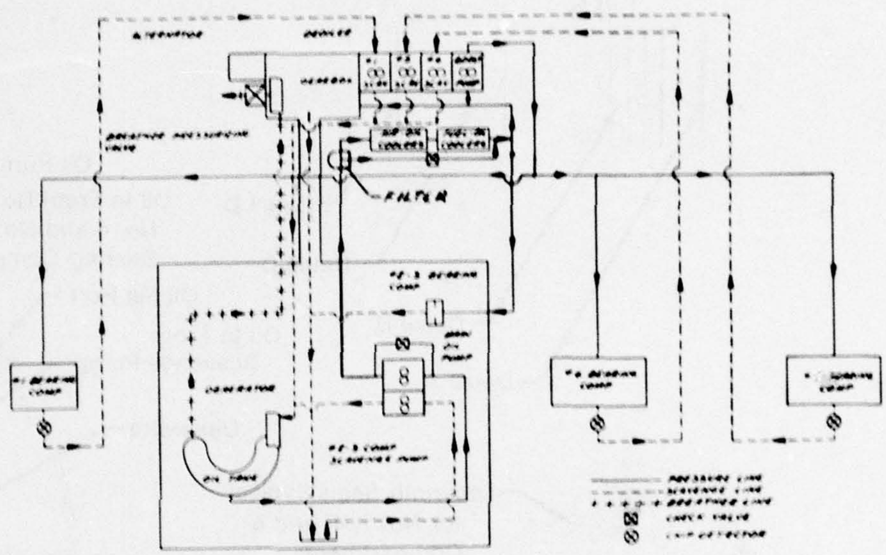


Figure 15. Final Compartmental Lubric





Flow Schematic

FD 101747

partmental Lubrication System Design (Sheet 1 of 2)

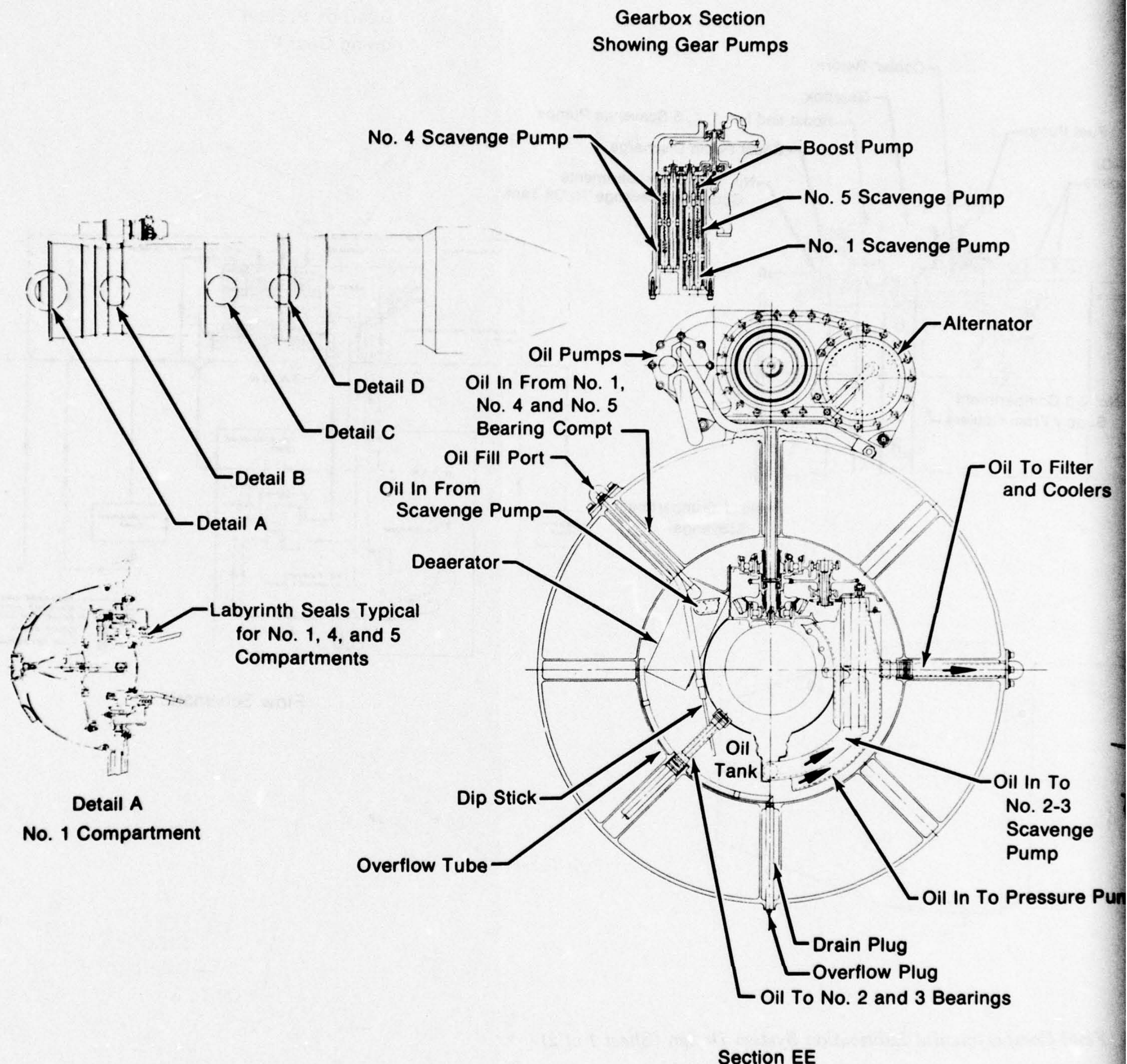
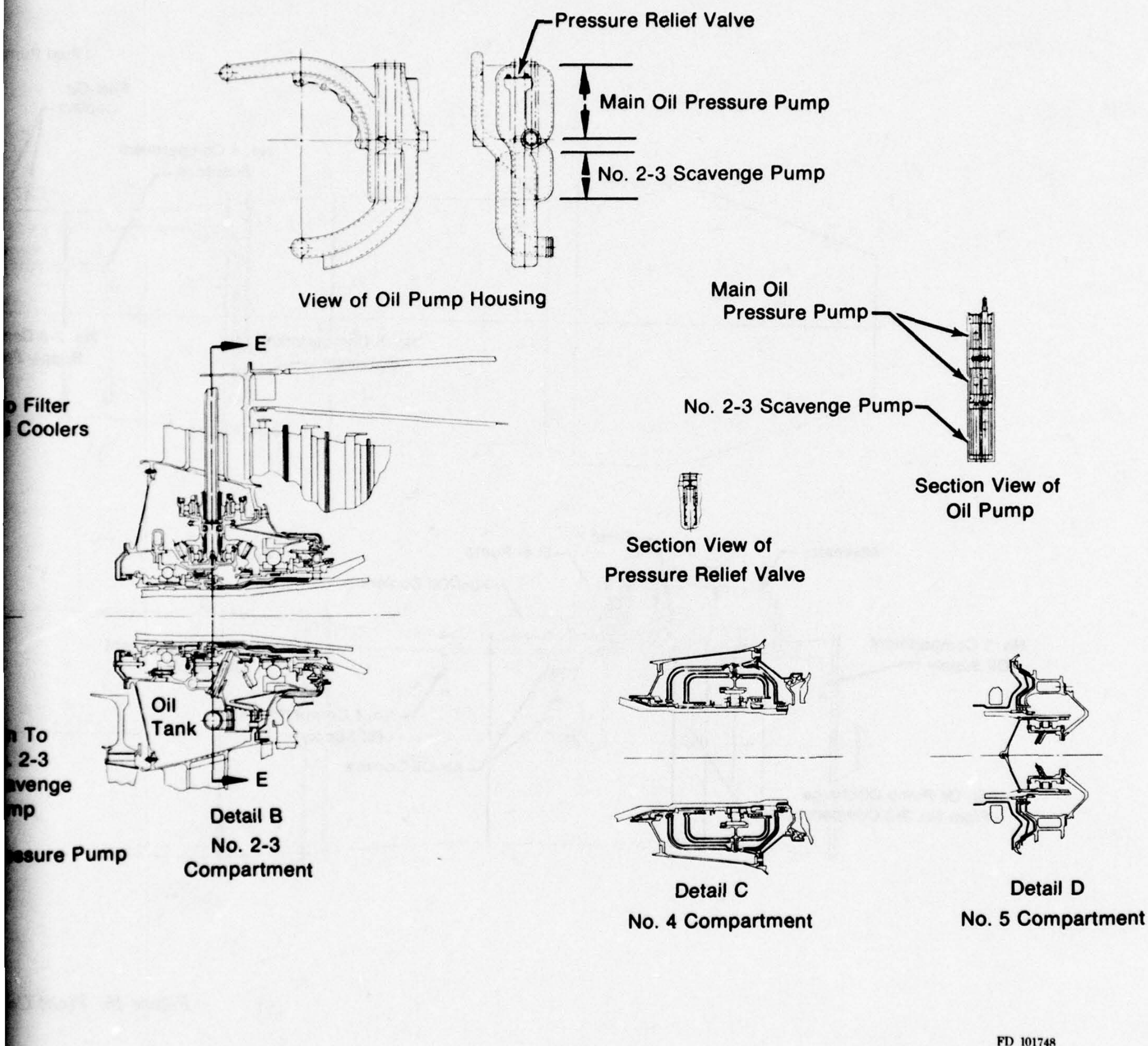


Figure 15. Final Compartmental Lubrication



ubrication System Design (Sheet 2 of 2)

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TABLE 8
QUANTITATIVE COMPARISONS OF SCAVENGE OPTIONS

Rating Criteria	*Vulnerability*	*Maintainability*	*Reliability*	*Acquisition Costs*	*Life-Cycle Costs*	*Weight*	*Frontal Area*	*Manufacturing Assembly and Development Difficulties*	*System Compromises*	*Totals*
Maximum Point Allotment	*30*	*25*	*10*	*5*	*5*	*10*	*8*	*3*	*4*	*100*
Scavenge Options										
I	30.0	11.2	9.5	5.0	5.0	10.0	8.0	1.8	0.7	81.2
II	29.1	10.8	9.0	4.7	4.9	9.7	7.9	1.5	1.0	78.6
III	27.9	10.0	8.9	4.8	4.8	9.7	7.9	1.3	0.9	76.2
IV	29.1	25.0	10.0	4.8	4.9	9.6	7.9	1.3	0.8	93.4
Baseline	20.7	18.4	8.9	4.6	4.7	9.7	7.5	3.0	4.0	81.5

NOTES:

1. Option I - Nos. 1, 4, and 5 compartments have scavenge blowdown and carbon seals.
2. Option II - Nos. 1, 4, and 5 compartments are scavenged by gearbox-mounted gear pumps. Carbon seals are used in compartments.
3. Option III - Nos. 1 and 5 compartments are scavenged by gear pumps mounted in their respective compartments. No. 4 scavenge pump is in gearbox. Carbon seals are used in compartments.
4. Option IV - Nos. 1, 4, and 5 compartments are scavenged by gearbox-mounted gear pumps. Labyrinth seals are used in the Nos. 1, 4, and 5 compartments and the volumetric displacement of the scavenge pumps used to limit seal leakage.

3. METHOD OF QUANTITATIVE ANALYSIS FOR PHASE II, TASK II EVALUATION

In Phase II, the selected compartment lubrication system configuration was compared with the baseline F100-PW-100 lubrication system on the basis of vulnerability, maintainability, reliability, weight, acquisition costs, life-cycle costs, frontal area, engine starting and windmilling operation, and oil contamination tolerance per the statement of work. Where applicable, the evaluation was done on a quantitative basis as a differential value (i.e., Δ weight, Δ cost, etc.) as compared to the baseline engine. The evaluations were made using the Phase II, Task I mechanical layout with component sizes substantiated by numerical analyses.

The method of evaluation for the vulnerability criteria was the same as applied in Phase I except for the weighting factors which were revised to better represent the vulnerability of the six views for a mixed mission. The revised weighting is shown below:

Views	*Weights, %*
Front	15
Rear	10
Top	15
Bottom	20
Left Side	20
Right Side	20

The method of evaluation for the maintainability, reliability, weight, acquisition costs, life-cycle costs, and frontal area criteria are identical to those used in Phase I except absolute differential values in comparison to baseline system are presented in Phase II. A point allotment system was used in Phase I.

The engine starting and windmilling operating was evaluated by comparing the compartmental lubrication system component parasitic power extraction to that of the baseline engines. The ability of the alternator to supply power during starting was also evaluated.

Oil contamination tolerance was evaluated by comparing running clearances of rotating parts, oil filtration capacity, and probability of inducing contamination into the compartmental lubrication system as compared to the baseline engine.

4. RESULTS OF PHASE II QUANTITATIVE ANALYSIS OF FINAL ENGINE DESIGN

Table 9 summarizes the results of the evaluation of the compartmental lubrication system in the areas of vulnerability, maintainability, reliability, acquisition costs, life-cycle costs, frontal area, engine starting and windmilling operation and oil contamination tolerance. All analyses were performed on a differential basis, compared with the baseline F100-PW-100 engine. Also, where practical, the analyses were performed quantitatively on a component basis to emphasize the compartmental lubrication system's strong and weak points. Details of the evaluations are given in the following paragraphs.

TABLE 9.
SUMMARY OF QUANTITATIVE ANALYSIS RESULTS

Criteria	*Criteria Differential Compared to Baseline F100-PW-100 Engine*
Vulnerability	Vulnerable area reduced 28.8%
Maintainability	Maintenance man-hours per million engine flight hours reduced by 5756
Reliability	Part discrepancies reduced by 962 per million engine flight hours
Weight	Weight increased by 1.7 lb
Acquisition Cost	Cost decreased by $906.00 per engine
Life-Cycle Cost	Life-cycle cost decreased by $4.1 million
Frontal Area	Frontal area decreased by 80 in.^{2}
Starting and Windmilling Operation	No change from baseline engine
Oil Contamination Tolerance	Time between filter cleaning reduced from 200 hours to 180 hours

a. Vulnerability

Table 10 shows on a component basis, the differential vulnerable areas of the selected system compared to the baseline engine for "A" and "B" kills of 30 to 50 cal projectiles traveling at 1500 and 2500 ft/sec. These numbers were then calculated as a percentage of the baseline engine, averaged for "A" and "B" kills and multiplied by the probability of a hit (view factor) for each of the six views. The "A" and "B" kill numbers, times their view factor, were then averaged together for each view and this number for each view was then added together to obtain an overall value of percentage of baseline vulnerable area. Table 11 shows that the vulnerable area of the selected system is 71.2 percent of the baseline system.

b. Maintainability

The results of the maintainability analysis are detailed to the component basis in Table 12 as differential maintenance man-hours per million engine flight hours (ΔMMH/10^6 EFH) compared to the baseline F100-PW-100 engine. The overall reduction in maintenance man-hours per million engine flight hours is 5756 ΔMMH/10^6 EFH. This reduction is primarily due to replacing carbon seals in the No. 1, 4, and 5 compartments with labyrinth seals.

TABLE 10
DIFFERENTIAL VULNERABLE AREA COMPARED TO BASELINE ENGINE

Component	View		ΔVulnerable Area (in.²)								Remarks
		Kill =	A	A	A	A	B	B	B	B	
		Cal. =	30	50	30	50	30	50	30	50	
		ft/sec =	1500	1500	2500	2500	1500	1500	2500	2500	
No. 1 Bearing Compartment	Front		0	−4.9	−2.9	−4.8	−19.6	−19.6	−14.7	−14.7	
	Rear		0	0	0	0	0	0	0	0	
No. 1 Bearing	Top		0	0	0	0	−2.2	−2.2	−2.2	−2.2	
	Bottom		0	0	0	0	−10.0	−10.0	−10.0	−10.0	
	Left Side		0	0	0	0	−5.6	−5.6	−5.6	−5.6	
	Right Side		0	0	0	0	−5.6	−5.6	−5.6	−5.6	
No. 2-3 Bearing Compartment	Front		0	0	0	0	0	0	0	0	
	Rear		0	0	0	0	0	0	0	0	
No. 2 Bearing	Top		−2.1	−11.9	−6.2	−10.4	−9.5	−15.7	−15.9	−26.5	
No. 3 Bearing	Bottom		+1.9	+11.6	+6.1	+10.2	+16.2	+27.0	+30.6	+51.0	
Bull Gear and Bevel Gear	Left Side		+1.0	+1.6	+0.5	+0.7	−30.3	−42.5	−34.1	−46.9	
	Right Side		+1.0	+1.6	+0.5	+0.7	−30.3	−42.5	−34.1	−46.9	
Towershaft (including portion in strut) Strut Area (12:00 position)											
No. 4 Bearing Compartment	Front		0	0	0	0	0	0	0	0	
	Rear		0	0	0	0	0	0	0	0	
No. 4 Bearing	Top		0	0	0	0	−7.2	−11.9	−10.2	−17.1	
	Bottom		0	0	0	0	−12.4	−20.7	−15.9	−26.5	All "A" kills are same as baseline
	Left Side		0	0	0	0	−6.2	−10.3	−8.8	−14.7	
	Right Side		0	0	0	0	−6.2	−10.3	−8.8	−14.7	
No. 5 Bearing Compartment	Top		0	0	0	0	0	0	0	0	
	Rear		0	0	0	0	−0.4	−0.4	−0.4	−0.4	
No. 5 Bearing	Top		0	0	0	0	−5.3	−5.3	−6.6	−6.6	
	Bottom		0	0	0	0	−12.8	−12.8	−14.6	−14.6	All "A" kills are same as baseline
	Left Side		0	0	0	0	−7.4	−7.4	−9.2	−9.2	
	Right Side		0	0	0	0	−7.4	−7.4	−9.2	−9.2	

TABLE 10
DIFFERENTIAL VULNERABLE AREA COMPARED TO BASELINE ENGINE (Continued)

			ΔVulnerable Area (in.²)								
		Kill =	*A*	*A*	*A*	*A*	*B*	*B*	*B*	*B*	
		Cal. =	*30*	*50*	*30*	*50*	*30*	*50*	*30*	*50*	
Component	*View*	*ft/sec =*	*1500*	*1500*	*2500*	*2500*	*1500*	*1500*	*2500*	*2500*	*Remarks*
Fuel-Oil Coolers	Front		−1.1	−0.2	−3.9	+0.7	−9.7	−10.6	−6.9	+11.5	
	Rear		−2.1	−2.1	−2.1	−2.1	−2.1	−2.1	−2.1	−2.1	
	Top		+24.2	+24.2	+24.2	+24.2	+24.2	+24.2	+24.2	+24.2	
	Bottom		−72.8	−72.8	−66.3	−62.0	−72.8	−72.8	−66.3	−62.8	
	Left Side		+2.9	−7.1	−0.9	−13.3	+2.9	−7.1	−0.9	−13.3	
	Right Side		−37.7	−37.7	−37.7	−37.7	−37.7	−37.7	−37.7	−37.7	
Plumbing	Front		0	0	0	0	−46.1	−46.1	−46.1	−46.1	
	Rear		0	0	0	0	+27.4	+27.4	+27.4	+27.4	
	Top		0	0	0	0	+63.4	+63.4	+63.4	+63.4	All "A" kills same as baseline
	Bottom		0	0	0	0	−196.8	−196.8	−196.8	−196.8	
	Left Side		0	0	0	0	−38.4	−38.4	−38.4	−38.4	
	Right Side		0	0	0	0	−67.2	−67.2	−67.2	−67.2	
Total	Front		−1.1	−5.1	−6.8	−4.1	−152.7	−152.7	−150.6	−146.6	
	Rear		−2.1	−2.1	−2.1	−2.2	−46.1	−46.1	−49.5	−50.7	
	Top		+39.9	+41.9	+35.8	+39.7	+69.5	+54.2	+67.1	+41.7	
	Bottom		−88.7	−90.5	−78.0	−81.4	−410.1	−370.3	−401.0	−396.8	
	Left Side		+1.9	−8.8	−0.8	−13.3	−73.6	−82.4	−87.1	−136.0	
	Right Side		−34.7	−132.8	−36.8	−36.3	−445.1	−447.3	−448.0	−449.1	

TABLE 10
DIFFERENTIAL VULNERABLE AREA COMPARED TO BASELINE ENGINE (Continued)

			Δ *Vulnerable Area (in.²)*								
		Kill =	*A*	*A*	*A*	*A*	*B*	*B*	*B*	*B*	
		Cal. =	*30*	*50*	*30*	*50*	*30*	*50*	*30*	*50*	
Component	*View*	*ft/sec =*	*1500*	*1500*	*2500*	*2500*	*1500*	*1500*	*2500*	*2500*	*Remarks*
Oil Tank	**Front**		0	0	0	0	−75.7	−75.7	−75.7	+75.7	
	Rear		0	0	0	0	−63.1	−63.1	−63.1	−63.1	
	Top		0	0	0	0	−12.7	−17.1	−14.9	−20.9	All "A" kills same as baseline
	Bottom		0	0	0	0	−51.1	−13.8	−35.1	+12.9	
	Left Side		0	0	0	0	+21.2	+35.3	+26.5	+9.6	
	Right Side		0	0	0	0	−185.8	−171.7	−180.5	−162.9	
Oil Filter	**Front**		0	0	0	0	+3.8	+3.8	+3.4	+3.2	
	Rear		0	0	0	0	0	0	−1.0	−0.5	
	Top		0	0	0	0	+9.3	+9.3	+9.3	+8.9	All "A" kills same as baseline
	Bottom		0	0	0	0	−4.9	−4.9	−4.9	−4.2	
	Left Side		0	0	0	0	0	0	−2.4	−3.9	
	Right Side		0	0	0	0	−8.6	−8.6	−8.6	−8.6	
Oil Pumps Supply and Scavenge	**Front**		0	0	0	0	−9.3	−9.3	−9.3	−9.3	All "A" kills same as baseline
	Rear		0	0	0	0	+7.6	+7.6	+5.2	+3.5	
	Top		0	0	0	0	+45.6	+45.6	+45.6	+40.4	"B" kill results from G/B hit first
	Bottom		0	0	0	0	−65.5	−65.5	−65.5	−65.5	"B" kill results from oil tank hit first
	Left Side		0	0	0	0	−4.3	−0.9	−2.6	+2.0	
	Right Side		0	0	0	0	−45.5	−45.5	−45.5	−45.5	
Main Gearbox	**Front**		0	0	0	0	−15.5	−15.5	−15.5	+15.5	
	Rear		0	0	0	0	−15.5	−15.5	−15.5	−15.5	
	Top		+17.8	+29.6	+17.8	+25.9	+31.4	+31.4	+41.9	+30.6	
	Bottom		−17.8	−29.6	−17.8	−29.6	−67.5	−67.5	−90.0	−84.8	
	Left Side		−2.0	−3.3	−0.4	−0.7	−5.5	−5.5	−11.6	−11.6	
	Right Side		+2.0	+3.3	+0.4	+0.7	−50.8	−50.8	−50.8	−50.8	
Air/Oil Coolers (4)	**Front**		0	0	0	0	0	0	0	0	
	Rear		0	0	0	0	0	0	0	0	
	Top		0	0	0	0	0	0	0	0	Same as baseline configuration
	Bottom		0	0	0	0	0	0	0	0	
	Left Side		0	0	0	0	0	0	0	0	
	Right Side		0	0	0	0	0	0	0	0	

TABLE 11
VULNERABILITY SUMMARY

View	*View Factor*	*Average % of Baseline Engine Vulnerable Area for "A" Kills*	*"A" Kill Average Times View Factor*	*Average % of Baseline Engine Vulnerable Area for "B" Kills*	*"B" Kill Average Times View Factor*	*"A" and "B" Kill Average With View Factor*
Front	15%	88.0	13.2	41.0	6.2	9.7
Rear	10%	89.0	8.9	79.5	8.0	8.5
Top	15%	138.0	20.7	111.3	16.7	18.7
Bottom	20%	28.5	5.7	41.8	8.4	7.1
Left Side	20%	97.3	14.6	83.3	16.7	15.7
Right Side	20%	66.8	13.4	47.8	9.6	11.5
Total	100%					$\Sigma = 71.2$

TABLE 12
MAINTAINABILITY ANALYSIS RESULTS

Component	Δ*MMH/10*^{6} EFH
Alternator	0
Gearbox	−170
Oil Filter	0
Oil Supply Pump	7904
No. 1 Scavenge Pump	−143
No. 2-3 Scavenge Pump	7061
No. 4 Scavenge Pump	−106
No. 5 Scavenge Pump	−143
Fuel Oil Coolers	0
Air-Oil Coolers	0
Boost Pump	0
Deaerator	65
No. 1 Bearing Compartment	−2544
No. 2-3 Bearing Compartment	1120
No. 4 Bearing Compartment	−13604
No. 5 Bearing Compartment	−5664
Oil Tank	900
Inlet Fan Module	−431
Total	−5756

c. Reliability

Reliability calculations detailed to the component basis are presented in Table 13. Reliability is expressed as the differential part discrepancies per million engine flight hours (Δpart discrepancies/10^6 EFH). The improvement in reliability is 962 fewer part discrepancies per million engine flight hours compared to the baseline engine.

d. Weight

Table 14 shows that the compartmental lubrication system results in a 1.7 lb increase in engine weight over the baseline engine. The increase in weight comes from operating two pump packages rather than one and providing a shielding cover for the alternator. This is partially compensated by a weight reduction realized from the compartmentalized oil tank and the use of labyrinth seals in place of carbon seals.

TABLE 13
RELIABILITY ANALYSIS RESULTS

Component	*Δ Part Discrepancies/ 10^6 EFH*
Alternator	0
Gearbox	−84
Oil Filter	0
Oil Supply Pump	99
No. 1 Scavenge Pump	11
No. 2-3 Scavenge Pump	88
No. 4 Scavenge Pump	11
No. 5 Scavenge Pump	11
Fuel-Oil Coolers	0
Air-Oil Coolers	0
Boost Pump	0
Deaerator	0
No. 1 Bearing Compartment	−240
No. 2-3 Bearing Compartment	35
No. 4 Bearing Compartment	−378
No. 5 Bearing Compartment	−320
Oil Tank	−195
Inlet Fan Module	0
Total	−962

TABLE 14
DIFFERENTIAL WEIGHTS OF COMPARTMENTAL LUBRICATION SYSTEM COMPARED TO BASELINE ENGINE

Source of Weight Differential	*Weight Differential (lb)*
No. 2-3 Bearing Compartment and Oil Tank	−5.6
Oil Supply and Scavenge Pumps	+5.1
Cover for Gearbox Mounted Alternator	+3.2
Bearing Compartment Seals (Labyrinth Seals Replace Carbon Seals)	−1.0
Total	+1.7

e. Acquisition Costs

Table 15 presents a component breakdown of the cost differential between the compartmental lubrication system and the baseline engine. The total cost savings would be $906 per engine obtained primarily from the internal oil tank and the use of labyrinth seals.

TABLE 15
DIFFERENTIAL ACQUISITION COSTS OF COMPARTMENTAL LUBRICATION SYSTEM COMPARED TO BASELINE ENGINE

Source of Acquisition Costs Differential	*Differential Costs Dollars*
Internal Oil Tank	−1476
Alternator Housing	+ 50
No. 2-3 Compartment	
Add: 3 Drive Gears	+ 195
Add: 2 Bearings and Bearing Housing	+ 175
Add: 1 Pump Housing	+ 175
Add: Pump Housing Support	+ 100
Add External Pump Housing	+ 505
Revise Main Pump Housing	− 130
Replace 4 Carbon Seals with Labyrinth Seals	− 500
Total	− 906

f. Life-Cycle Costs

Table 16 shows that the compartmental lubrication system would result in a 4.1 million dollar reduction in the life-cycle costs. Using labyrinth seals in the No. 1, 4, and 5 compartments accounts for 2.9 million of the 4.1 million dollar total.

TABLE 16
DIFFERENTIAL LIFE CYCLE COSTS OF COMPARTMENTAL LUBRICATION SYSTEM COMPARED TO BASELINE ENGINE

Source of Life Cycle Cost (LCC) Differential	*Differential LCC - $ Millions*
No. 2-3 Bearing Compartment and Scavenge Revisions	+0.6
Oil Supply and No. 2-3 Scavenge Pumps in No. 2-3 Compartment	+2.1
Oil Tank in No. 2-3 Compartment	−2.5
Gearbox on Top of Engine	−0.2
Fuel-Oil Coolers on Top of Engine	0
Plumbing Revisions	−1.2
Labyrinth Seals in No. 1 Compartment	−0.5
Labyrinth Seals in No. 4 Compartment	−1.5
Labyrinth Seals in No. 5 Compartment	−0.9
Total	−4.1

g. Frontal Area

The frontal area of the compartmental lubrication system was found to be only 80 square inches less than the baseline F100-PW-100 engine. This area reduction was primarily due to moving the oil tank inside the No. 2-3 bearing compartment. Changes in oil plumbing had little effect on the projected frontal area since most of the oil plumbing was either hidden by or in the same plane as the fuel plumbing.

h. Engine Starting and Windmilling Operation

During Phase I of this project, an attempt was made to mount the alternator in the No. 1 compartment to satisfy a statement-of-work requirement that the lubrication system design shall provide as an option for internal location of the engine alternator. Other internal engine locations such as the No. 2-3 and No. 5 compartments were ruled out due to insufficient space or hot environment.

A Phase II study showed that while cranking the engine at the minimum high rotor lightoff speed of 3000 rpm, the lower rotor turns at 300 to 400 rpm, well below the speed required by the alternator to provide adequate energy to the main combustor ignition system. Several optional methods of starting the engine were investigated such as batteries and jet fuel starter powered generators, each of which would result in an excessive weight, cost, and maintainability penalty. The problem was reviewed with the AFAPL Project Engineer in correspondence dated 26 May 1976. The alternator was moved back to the gearbox location for the selected system. Subsequent quantitative analysis has shown that the bullet nose location did not offer any improvement in vulnerability, and resulted in a slight increase in cost and weight.

With the alternator relocated on the gearbox like the baseline engine, the starting and windmilling operation of the compartmental lubrication system engine is essentially the same as that of the baseline engine. If the blowdown system had proven desirable in the quantitative evaluation, a slight reduction in parasitic losses (less than 3.5 hp at full power) would have been realized through the elimination of the oil boost pump and No. 1, 4, and 5 compartment scavenge pumps. However, under the present configuration, the power requirements of the selected system are the same as that of the baseline engine.

i. Oil Contamination Tolerance

Bearing compartment air leakage for the compartmental lubrication system is two to three times that of the baseline F100-PW-100 engine due to the use of labyrinth seals in place of carbon seals. Consequently, the lubrication system contamination due to air leakage will increase proportionately. However, it is estimated that in the F100-PW-100 engine, air leakage only accounts for 10 percent of the oil system contamination due to the judicial selection of clean seal pressurization air. Air to pressurize the rear of the No. 2-3 compartment and No. 5 compartment is bled inward from the compressor sixth stage to the engine bore. The heavier contamination particles are thus held at the engine OD flow path providing clean air at the engine bore to pressurize the seals. The front of the No. 2-3 compartment and the No. 1 compartment are similarly pressurized using fan discharge air. Seal pressurization air for the No. 4 compartment is bled from the engine OD at the seventh compressor stage. However, this air is passed through a centrifugal filter which removes 92 percent of the contaminants before it is used to pressurize the No. 4 seals. A seal pressurization system similar to this would have to be used on the compartmental lubrication system engine to minimize the contamination problem with labyrinth seals. It is estimated that the time between cleaning for the oil filter will be reduced 10 percent from 200 hours to 180 hours due to the use of the labyrinth seals.

The 10,000 rpm oil pumps for the compartmental lubrication system will have roughly the same clearances, gear tip speed, and bearing loads as the baseline engine. However, the bearing speed has increased 40 percent over the baseline engine. The remainder of the lubrication system, as expected, is similar to the baseline engine in contamination tolerance.

SECTION IV
PHASE III — DETAILED DESIGN AND BENCH TEST

1. SUBSYSTEM DESIGN ANALYSIS

a. General

This section provides the design criteria and approach that was used for the design of critical components and rig hardware required for test substantiation of the selected compartmental lubrication system. The trade studies and preliminary design efforts of Phases I and II resulted in a compartmental lubrication system which achieved reduced vulnerability through location of major lubrication system components in the largest bearing compartment (No. 2-3).

Critical items identified from the selected system, as requiring design and test substantiation, were the high-speed oil supply and scavenge pumps (two and one-half times the speed of conventional engine pumps), associated high-speed drive gear train, a small volume oil tank, and capability for tank deaeration of labyrinth seal leakages in excess of three times that of conventional engines. The high-speed oil supply and scavenge pumps plus the small volume oil tank were designed during the critical component design task (Phase III, Task I). These components were fabricated and successfully bench tested during Phase III, Task II to qualify them for the system rig.

The F100-PW-100 No. 2-3 compartment rig (F34024) (Figures 16 and 17) was selected for the system rig tests. Utilization of this existing rig with modifications for incorporating high-speed oil pumps, associated high-speed drive train, a small volume oil tank, and deaeration system within the compartment provided an effective system test bed at a minimum cost.

The No. 2-3 compartment rig has the capabilities for simulating engine speeds, compartment altitude environmental conditions, internal pressures, temperatures, and required oil flowrates. This provided an ideal test fixture for evaluation of the compartmental lubrication system concept.

The feasibility of running the No. 2-3 compartment rig inverted was investigated. This would have provided for the towershaft and high-speed pump drive train at the top of the compartment allowing space for an integral oil tank at the bottom of the compartment. This scheme was found to be feasible, but the advantages did not outweigh the $39,953 cost for the additional rig modifications. It was necessary to design a self-contained oil tank to keep oil off the towershaft gears, but a tank of this type was already required for the component bench tests.

b. Description of Test Articles

(1) Oil Supply and Scavenge Pumps

The compartmental lubrication system oil supply and No. 2-3 compartment scavenge pumps (Figure 18) were designed to operate at a full power speed of 10,000 rpm (two and one-half times that of conventional engine oil pumps) to provide a smaller vulnerable area and to reduce the size of the gear train required to drive the pump. The pump supply and scavenge elements are stacked in a common housing and are driven off the towershaft bevel gear by spur gears. Pump gears are of a 9 tooth-16 pitch (pitch = No. of teeth/pitch diameter) configuration to provide adequate capacity without exceeding pump tip speed cavitation limits at the high shaft speeds.

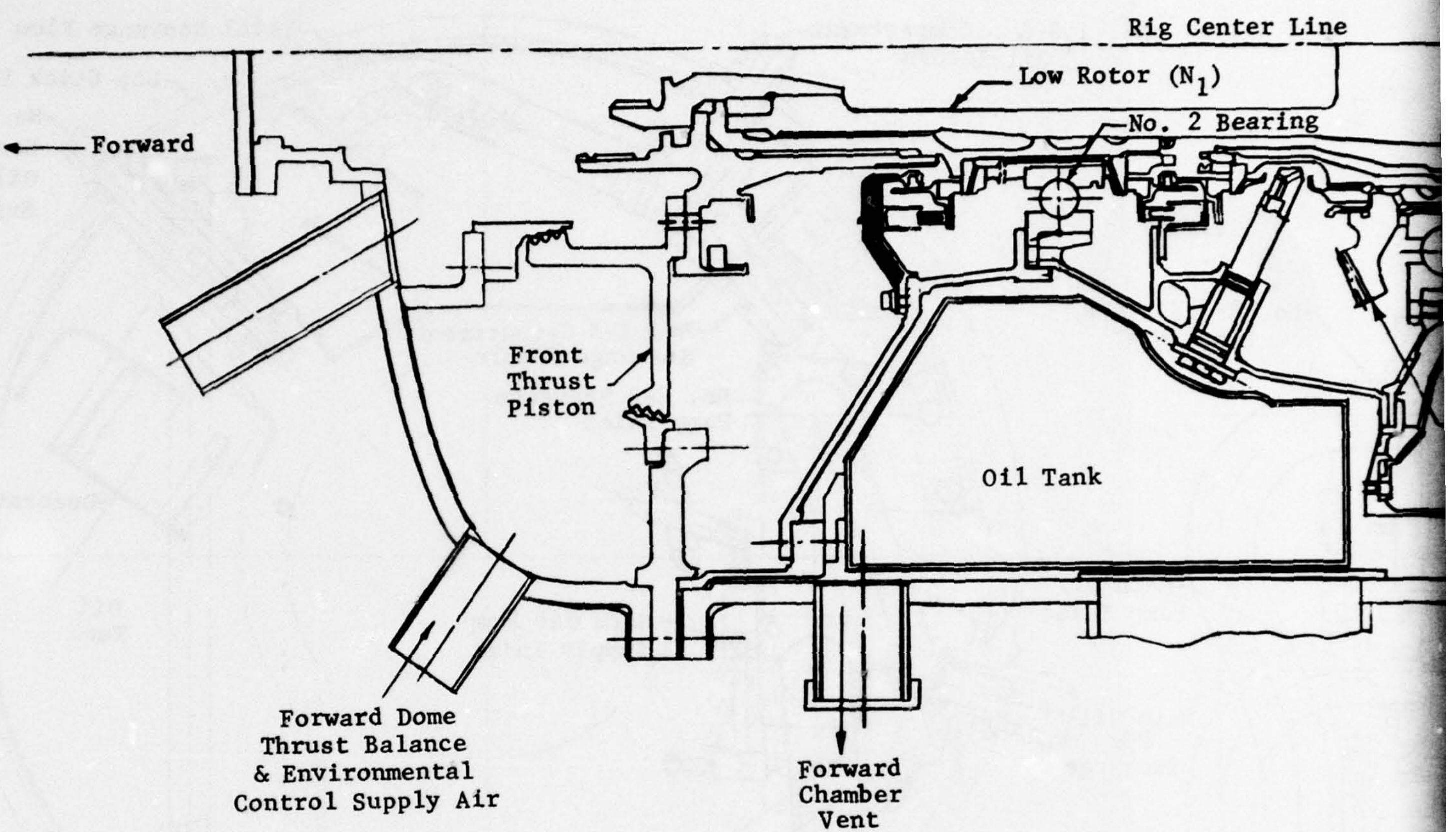

Figure 16. Compartmental Lubrication System N
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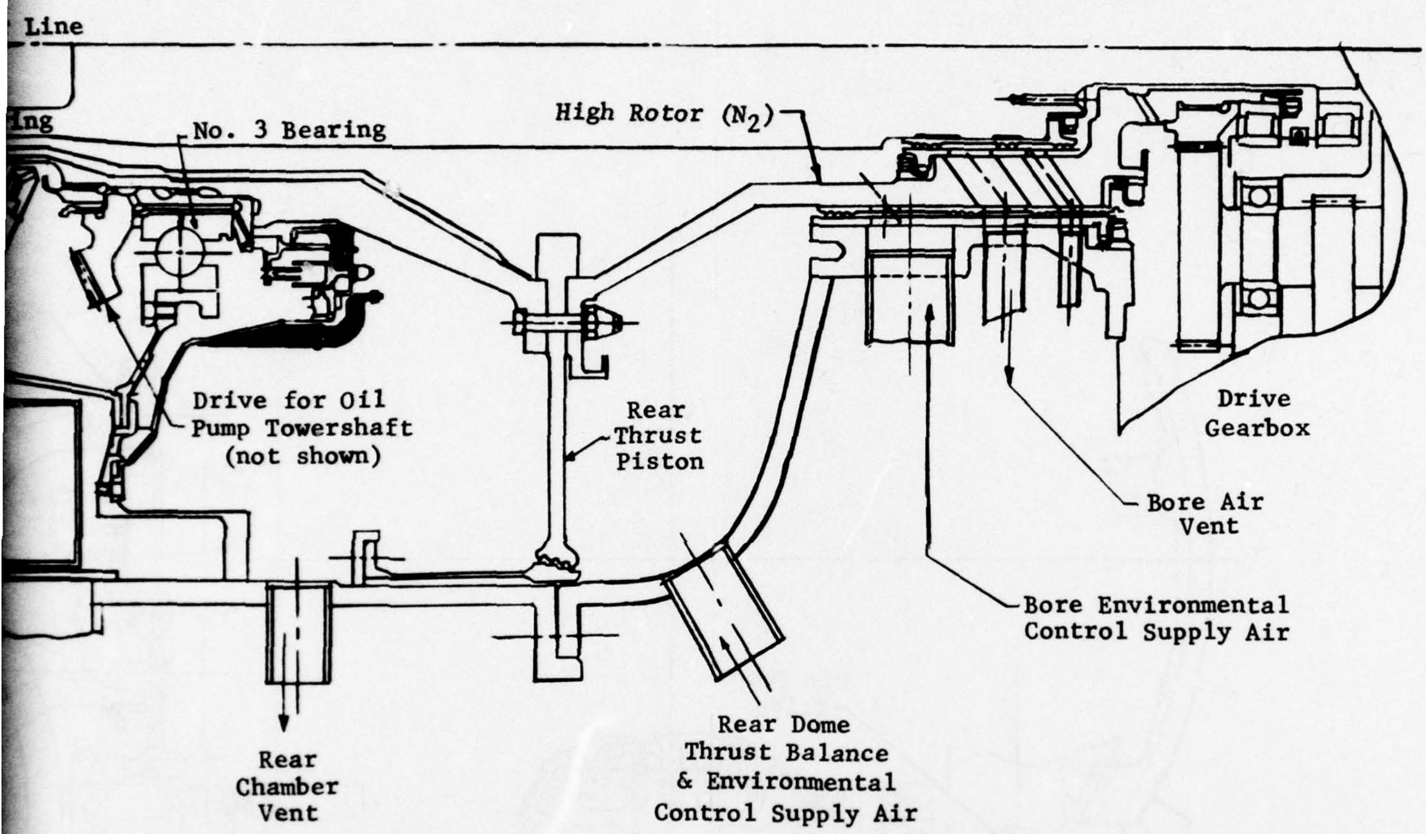

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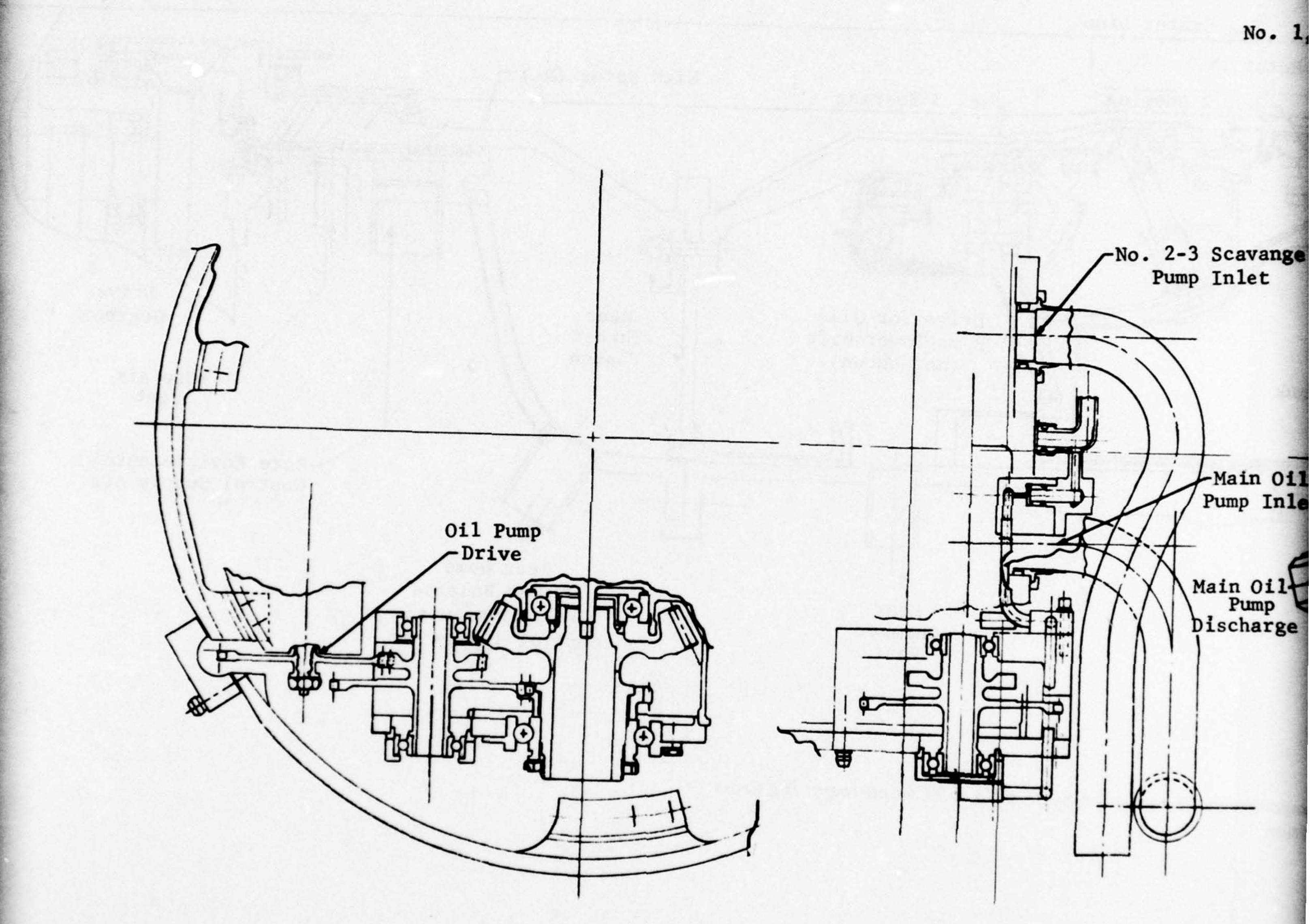

Figure 17. Arrangement of Drive Train, Pump, an ment Rig

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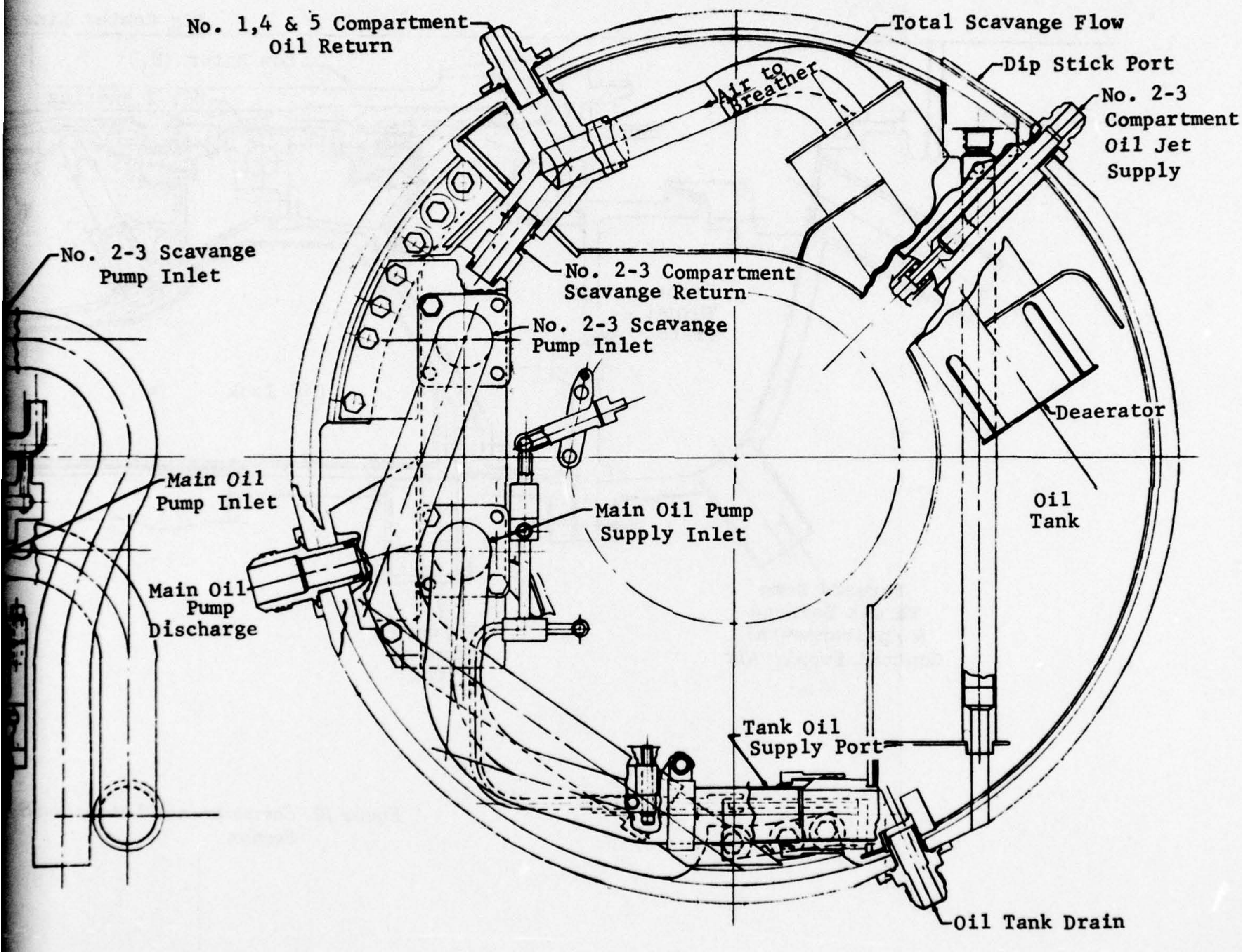

t of Drive Train, Pump, and Tank in No. 2-3 Compart-

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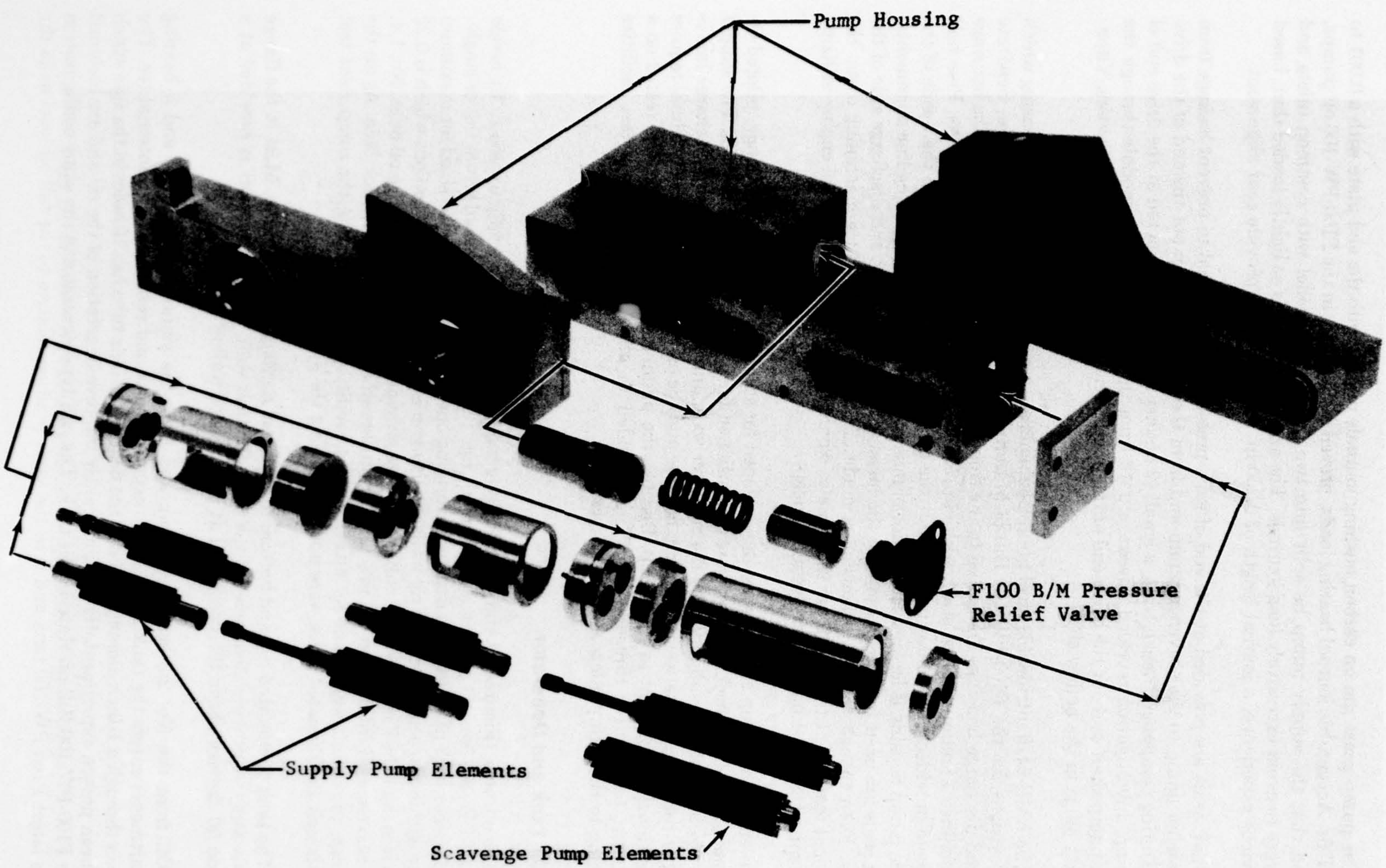

Figure 18. High-Speed Oil Pump Internal Parts

The pump gears run on carbon bushing journals pressed into the end plate with a 0.001 to 0.0025T fit. Acceptable journal bearing loads, proven satisfactory in the F100-PW-100 oil pumps, required that the supply pump be split into two pumps in parallel with common inlets and discharges to avoid excessively long journals. The scavenge pump is so lightly loaded that, based on previous experience, a journal length of 0.250 in. was chosen to provide gear alignment.

Shaft seals are provided at the end of each pressure stage journal to prevent leakage from the pressure pump to the scavenge pump and from the pressure pump out the end of the drive shaft. During component bench tests, a small (2 qts/hr) shaft leak was noted at the drive end of the pump. Investigation revealed a leakage path from the scavenge pump module through the supply pump shaft and out the drive end of the pump. The leak was stopped by inserting Viton-A gasket plugs in the hollow shaft.

The AMS-6470 nitrideable steel pump elements are stacked in an aluminum housing which has provisions for the F100-PW-100 Bill-of-Material cold start pressure relief valve. Separate inlet and discharge housings are bolted to the front and rear of the pump housing. The housings are machined aluminum plate stock with simple, straight cuts for cost-effectiveness. Five bolt holes were provided in the housing to mount the pump to an existing flange on the inside of the rig. The pump housing is located on two dowel pins to ensure gear and plumbing line alignments. O-ring seals are used in grooves between the housings to seal the inlet and discharge side of the pumps. Even though the pump housing is made of plate stock to reduce machining costs, the functional features of the pump are the same as would be required for an actual engine utilizing the Compartmental Lubrication System concept.

Inlets to the pump elements are dimensioned for standard 4-bolt, swivel-flange, piloted O-ring connectors. The discharge ports are dimensioned for piloted O-ring connectors. The supply pump discharge line is supported by the rig outer wall, and the scavenge pump discharge line is a jumper-tube trapped between the pump housing and the oil tank fitting. These two lines require no additional attachment to the pump. Because the pump was also designed to be tested on a component test bench, threaded holes are provided to allow the hook up of test facilities plumbing to the pump inlets and discharges.

(2) Oil Tank and Deaerator

The oil tank (Figures 19 and 20) was designed in a semicircular configuration to fit inside the F100-PW-100 No. 2-3 bearing compartment rig. The outer walls are made from flat or single-curved sheet metal parts to avoid expensive forming operations. The tank was sized for maximum volume within the confines of the rig walls resulting in a fill capacity of 2.75 gallons which is 0.25 gallon less than the F100-PW-100 tank. Bosses are provided for No. 2-3 scavenge oil inlet, No. 1,4, and 5 scavenge oil inlet, main oil out, tank drain, breather port, and a dipstick hole. As on the oil pump, provisions were made for attaching test facility plumbing during the component test even though the threaded holes were not used during the system tests.

The tank is mounted top and bottom to the same rig flange as the pump. Most of the flange was cut away to provide clearance for the oil tank outer wall. Another support is provided at a location 20 degrees above the horizontal at the forward wall of the tank.

Oil from the No. 2-3 scavenge pump and from the simulated No. 1, 4, and 5 bearing compartments enters the tank through two separate ports and combines in an internal tee. The oil flows through a 1-inch diameter tube to the deaerator. This tube has 10 holes in the side which have been proven experimentally to significantly improve deaeration of the oil and are included in the F100-PW-100 Bill-of-Material oil tank. The can type deaerator is the same configuration as the latest F100-PW-100 tank and is in the same position relative to the full oil level as in the

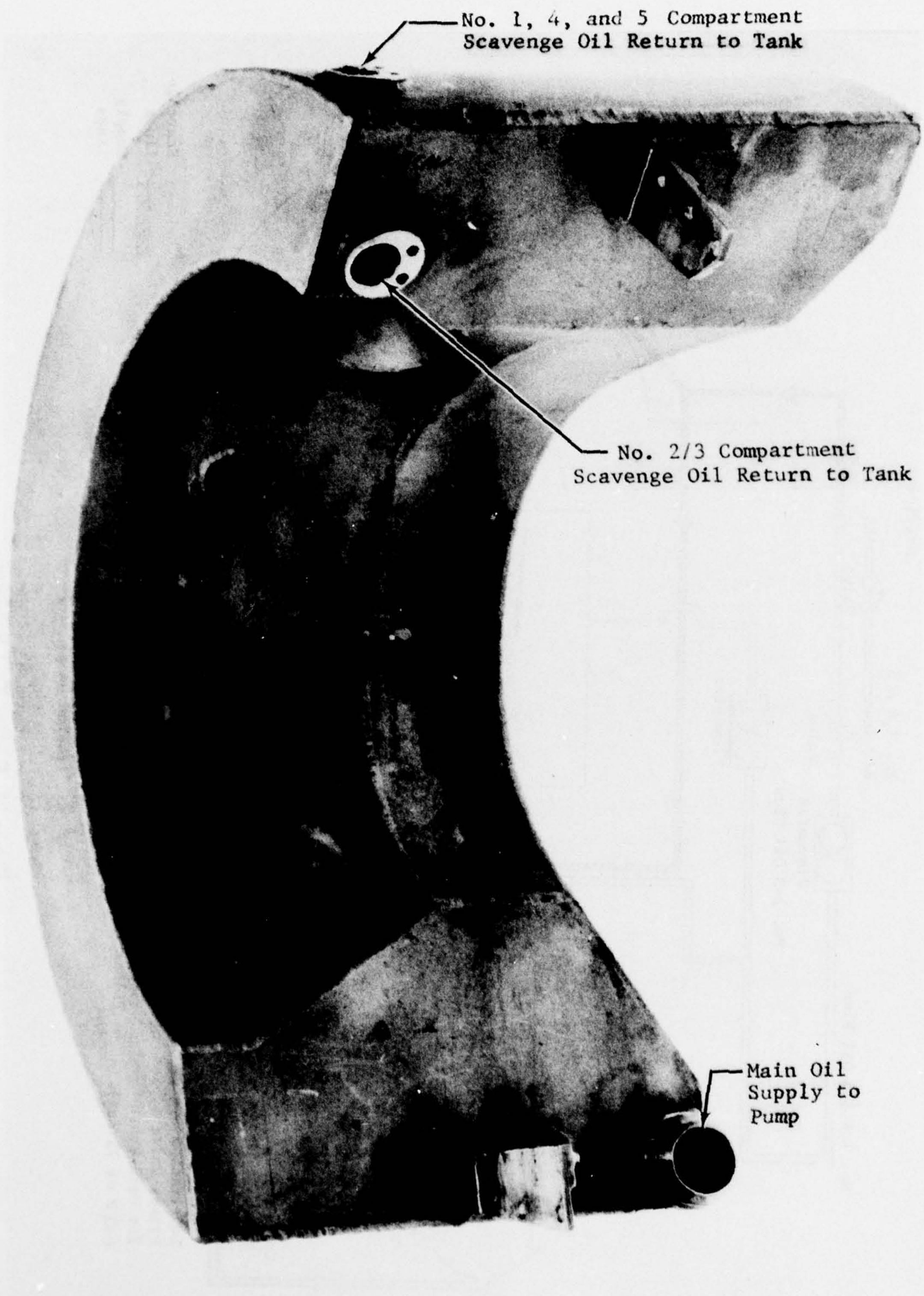

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Figure 19. Compartmental Lubrication System Oil Tank

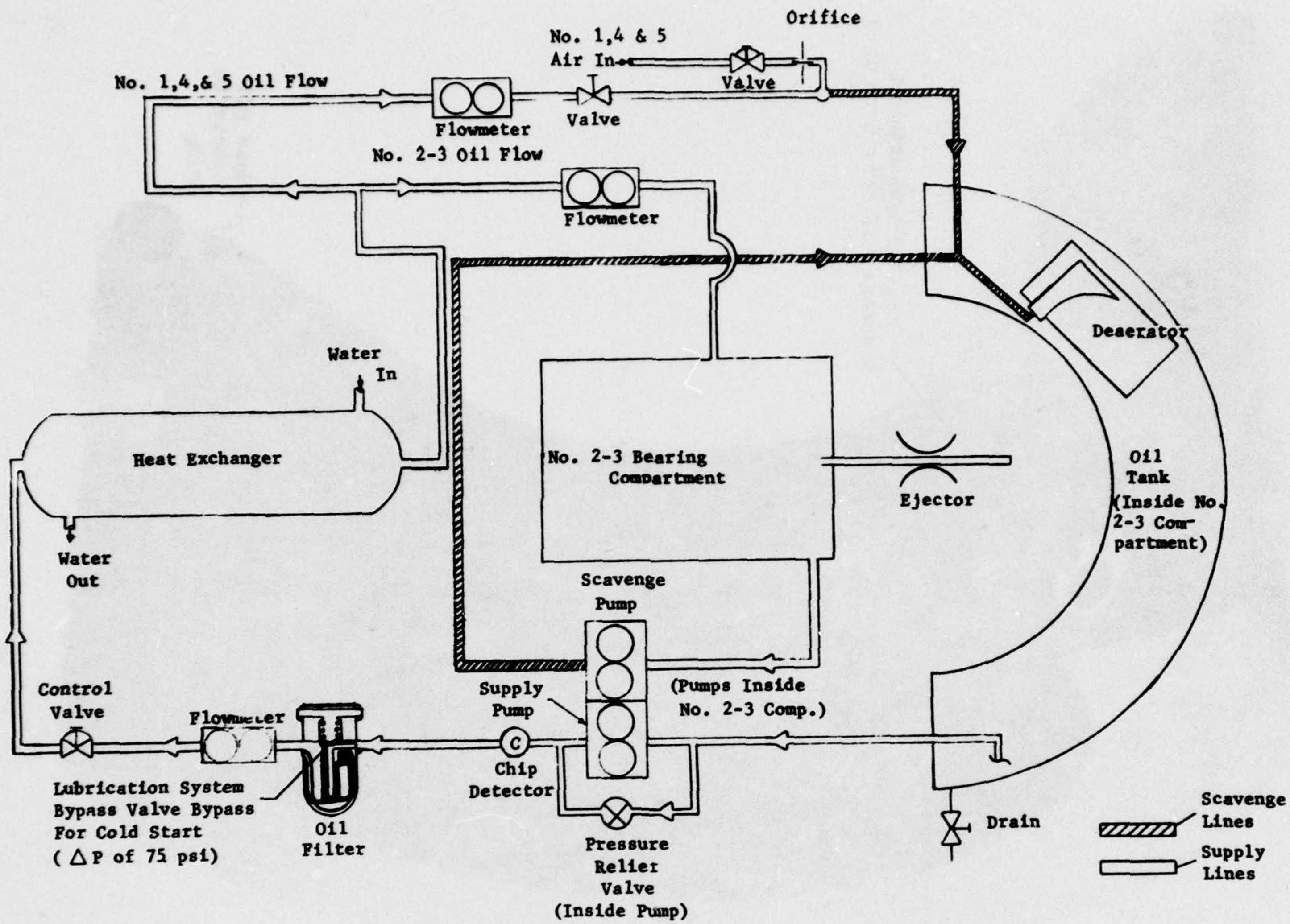

Figure 20. System Test Schematic

Bill-of-Material tank. The air and oil mixture is fed into the cylindrical deaerator tangentially at the top. As the mixture circulates in the cylinder, centrifugal force separates the lighter air from the oil. The air is fed out the top of the deaerator to the breather system while the oil drops to the bottom of the tank.

The tank has a modified AN-type connector for connecting the main supply pump plumbing, and a drain boss is provided to drain the tank while it is installed in the rig. A dipstick, calibrated after tank fabrication was completed to allow for tolerances on the sheet metal walls, was provided to check the tank oil level. The dipstick was not left in the tank during operation but was inserted by the test operator through a port in the rig outer case.

(3) System Test Rig

The system tests were conducted utilizing the F100-PW-100 No. 2-3 bearing compartment rig (No. F34024) as the test vehicle. Arrangement of the pumps, drive train, and tank within the rig is shown in Figure 18. This rig was designed to test a Bill-of-Material No. 2-3 compartment and towershaft at conditions simulating a typical fighter mission. The environment was matched to flight conditions by controlling the air pressure and temperature around the compartment as well as the oil supply temperature and rig speed. The thrust loads on the No. 2 and 3 bearings were controlled by thrust balance pistons at the front and rear of the compartment. Both the high and low rotors were driven off a single coaxial gearbox mounted on the rig.

This rig was modified for the Compartmental Lubrication System Tests to accommodate an internal oil tank, supply and scavenge pumps, and a gear train to drive the pumps. The front bearing support was redesigned to accommodate the oil tank. The No. 2-3 cross over support was redesigned to accommodate the pump and to allow space for the pump drive system. The front support ring was partially removed to provide for oil tank volume. The remaining portion of the ring was found to be sufficient to mount the oil pump package and drive system. A ring was designed for the front ring flange to mount the tank and front support. Access holes were provided through the rig for oil fill and level indications in the internal tank. The tests were conducted without a gearbox. The only towershaft power extraction was for the pump drive system.

A flow schematic for the system test rig is presented in Figure 20. Oil from the tank supply port was routed to the high-speed pump inlet through an internal rig line. Pump discharge flow was fed out of the rig and through a magnetic chip detector and 70 micron filter. A flow bypass valve in the pump provided for pressure relief above 175 psid. Downstream of the filter, the oil was routed through a flowmeter and then through a stand mounted shell and tube heat exchanger which was used to control the temperature of the oil supplied to the rig.

Approximately half of the oil was supplied to the rig oil jets. The remainder was bypassed and sent to the No. 1, 4, and 5 compartment oil tank inlet after having air simulating No. 1, 4, and 5 compartment labyrinth seal air leakage mixed with the oil. Oil fed to the No. 2-3 compartment jets was gravity drained to the bottom of the compartment after cooling and lubricating the bearings, seals, and gears. The scavenge element pumped oil from the bottom of the compartment and transported it to the oil tank where it was combined with No. 1, 4, and 5 compartment flow in an internal tee before entering the tank deaerator. After being deaerated, the oil dropped to the bottom of the tank where it was again supplied to the system. Air separated from the oil was vented out the top of the rig to a stand-mounted breather tank. Because this rig did not have a deoiler and breather valve system, any oil mist that settled to the bottom of the breather tank was returned to the oil tank by a stand-mounted low capacity pump. The test stand also had an ejector system which reduced breather pressure to simulate altitude operating conditions.

c. Design Criteria

(1) System Pressures

Maximum and minimum allowable design values for oil supply pressure and breather pressure are presented in Table 17. During the system tests, supply oil pressures were allowed to fall out based on preset oil flowrates but did not exceed the limits shown.

TABLE 17
SYSTEM OIL SUPPLY AND BREATHER PRESSURE

	Maximum	*Minimum*
Oil Pump Pressure Rise (psid)	195	40
No. 2-3 Oil Supply Relative to Breather (psid)	80	10
Breather Pressure (psia)	30	3

Breather pressures and rig environmental pressures which were set for the system tests are given in Figure 21.

(2) Oil Flowrates

Oil jets for the bearings were sized to provide lubrication and cooling and thus maintain bearing clearance. A design criterion of 100°F differential between oil supply and race temperature was used. Seal oil flows were sized to maintain acceptable seal temperatures while limiting mechanical churning heat generation. All gears were mist lubricated. Overall compartment temperature rise was limited to 100°F. A summary of oil flow to each component jet at sea level intermediate power conditions is given in Table 18. Total compartment oil flows for the test mission points are given in Figure 21. Individual jet flows for the mission points are in proportion to the sea level intermediate values. Oil type used for the test was MIL-L-7808G.

TABLE 18
SYSTEM RIG OIL JET FLOWS
AT SEA LEVEL INTERMEDIATE POWER

Jet Location	*No. of Jets*	*Flow Per Jet (ppm)*
No. 2 Front Seal Plate	1	4.5 to 6.0
No. 2 Bearing and Rear Seal Plate	1	16.0 to 19.0
No. 3 Front Seal Plate	3	2.0 to 3.0
No. 3 Bearing and Rear Seal Plate	3	11.0 to 13.0
Upper Towershaft Ball Bearing	1	0.5 to 1.5
Lower Towershaft Ball Bearing	1	0.5 to 1.5
Upper Idler Gear Ball Bearing	1	0.5 to 1.5
Lower Idler Gear Ball Bearing	1	0.5 to 1.5
	Totals	61.5 to 79.0

An internal manifold was designed to provide adequate cooling for the oil pump high-speed gear train bearings. The manifold, tapped into the main bearing supply jet, distributed oil to each idler shaft bearing and to the lower towershaft bearing. The upper towershaft bearing was lubricated by existing oil jets in the No. 2 bearing support. The minimum allowable jet size was

Flight Point	Condition	Time at Point, Min	Oil Supply Temperature, °F	Rotor Speed High N_2, rpm	Rotor Speed Low N_1, rpm	Bearing Loads No. 2 Bearing, lb	Bearing Loads No. 3 Bearing lb	Breathe Pressur psia
1	Sea Level Idle	15	207	9140	6581	848	1503	14.7
2	Climb	3	192	13009	9367	5243	9291	12.2
3	Cruise Out	31	251	10912	7857	1721	3050	6.8
4	Combat	5	196	12909	9295	4181	7410	8.3
5	Cruise Back	27	251	10912	7857	1721	3050	6.8
6	Sea Level Idle	16	233	9140	6581	848	1503	14.7

Total = 97 Minutes (31 Cycles Required for 50 Hours of Test)

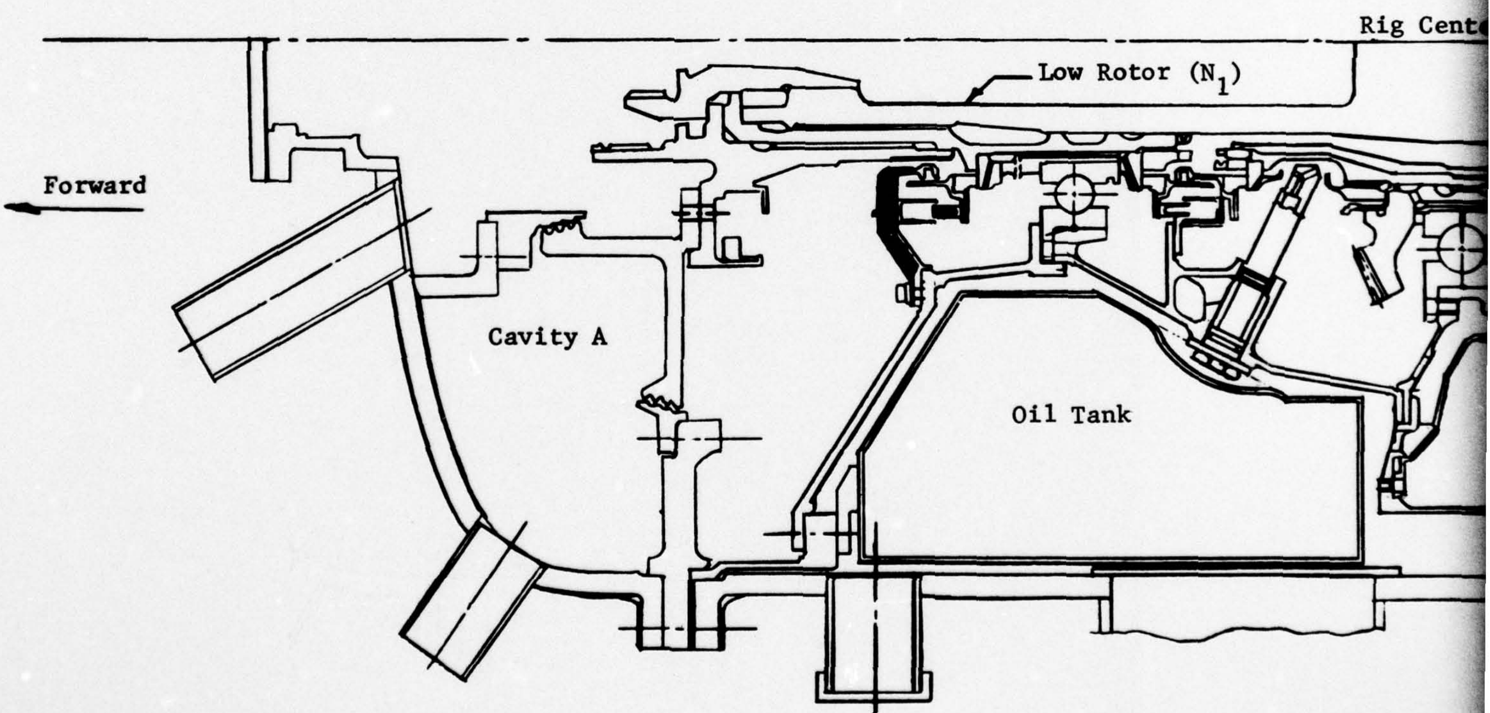

Figure 21. System Test

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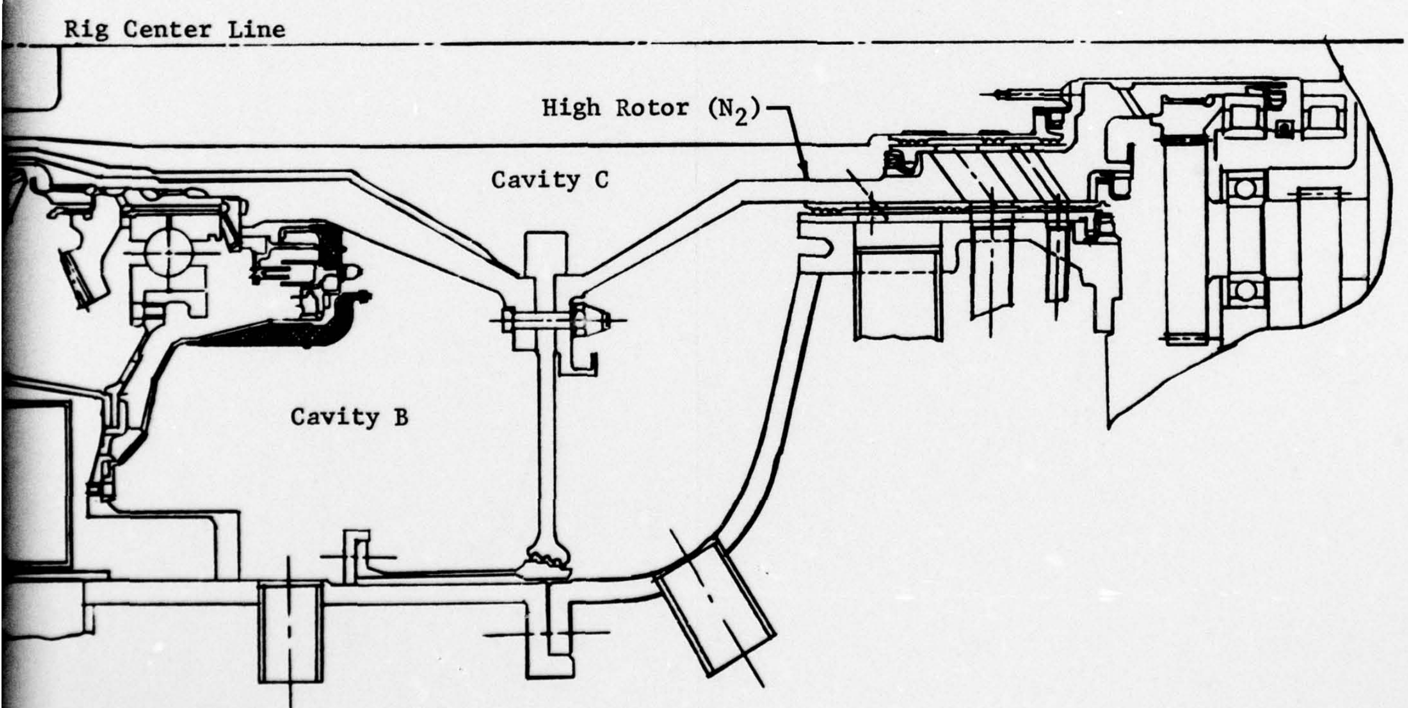
. 3 aring lb	Breather Pressure, psia	Compartment Pressures and Temperatures: Cavity "A" Pressure, psia	Cavity "A" Temperature, °F	Cavity "B" & "C" Pressure, psia	Cavity "B" & "C" Temperature, °F	Simulated No. 1,4,&5 Compartment Seal Air Leakage pph	No.2-3 Compt. Oil Flow Rate ppm	Est. No. 2-3 Compt Oil Temp. Rise (Supply to Discharge) °F
03	14.7	15	136	18	195	46	56	30±5
91	12.2	36	429	69	588	200	80	90±10
50	6.8	18	234	25	349	56.6	67	38±5
10	8.3	29	408	55	568	160	79	82±10
50	6.8	18	234	25	349	56.6	67	38±5
03	14.7	15	136	18	195	46	56	30±5



21. System Test Conditions

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set at 0.035-in. diameter to prevent particulates from blocking the jets. This required that an orifice be provided at the inlet to the manifold to reduce the pressure drop across each of the three supply jets. The oil was directed on the bearings in the direction each bearing is pumping oil (a thrust-loaded bearing pumps air and oil in the direction of thrust on the outer race).

(3) Deaeration Requirements

During the preliminary design phase, an analysis was conducted in which labyrinth seals were used in the No. 1, 4, and 5 compartments in conjunction with scavenge pumps sized to minimize air leakage and to prevent compartmental oil loss during engine deceleration. The analysis indicated that this scavenge breather system was practical from both an air leakage and an oil retention standpoint. Seal leakages, which were oil tank deaerated, were over three times that of conventional engines, but component bench tests conducted during Phase III, Task II have demonstrated the capability to deaerate this quantity of air with the can deaerator tank. No. 1, 4, and 5 compartment air flowrates which were deaerated in the system rig at the mission test points are tabulated on Figure 21. The values shown on Figure 21 reflect the use of labyrinth seals in the No. 1, 4, and 5 compartments.

(4) Rig Speed

Maximum design rig speeds are 13,900 rpm for the high rotor, 10,008 rpm for the low rotor, 26,702 rpm for the towershaft, and 10,000 rpm for the oil pump. Main shaft and towershaft speeds were selected to correspond with the F100-PW-100 engine values. High and low rotor speeds for mission points are tabulated in Figure 21. The rig coaxial gearbox drives both main shafts at a fixed gear ratio. The low rotor speeds were obtained by setting high rotor speed and applying a fixed gear ratio. A trip signal is provided on the drive to limit rig overspeed on the high-pressure rotor to 14,000 rpm.

(5) Temperatures

Oil scavenge temperatures were maintained below 300°F for all mission points. Maximum oil supply temperature was 251°F. Environmental air temperature was a maximum of 429°F in the front cavity and 588°F in the rear cavity. These temperatures correspond to the F100-PW-100 values at the selected mission points. Oil and air temperatures for each mission point are tabulated in Figure 21.

(6) Structural Limitations

Short time allowable material stress limits were set at:

- Bending Stress — 1. × 0.2 percent Yield at temperature
- Tensile Stress — 1. × 0.2 percent Yield at temperature
- For the Oil Tank:

  Buckling Factor of Safety ≥ 4.0
  Bending Stress Factor of Safety ≥ 3.0
  No creep problems because the maximum temperature was 300°F.

(7) Drive Gear Alignment

The JT9D main gearbox gear shapes and bearings were used for the pump drive train because they provided the required speed ratio. Consequently, tolerances on bearing fits and location of bearing housings were patterned after the JT9D main gearbox. Dowel pins or pilot diameters were used to ensure accurate alignment of mating parts. Tolerances were stacked for the towershaft-to-idler shaft mesh and for the idler shaft-to-pump gear mesh and then input into the spur gear design program (PWA Computer Program No. 5905) to determine tooth thickness reduction requirements.

Table 19 shows the tolerance stack for the towershaft-to-idler and idler-to-pump meshes along with the required and JT9D tooth thickness reductions. These values show that the two drive train gear meshes could accept additional tolerance stack without danger of binding.

TABLE 19
DRIVE TRAIN TOLERANCES

Mesh	*Tolerance Stack-up (in.)*	*Required Tooth Thickness Reduction (in.)*	*JT9D Gear Tooth Thickness Reduction (in.)*
Towershaft-to-Idler	±0.0090	0.004 to 0.008	0.0055 to 0.0095
Idler-to-Pump	±0.0138	0.007 to 0.011	0.0075 to 0.0115

(8) Gear Pump Tip Speed Limitations

Gear tip speeds were maintained below 30 ft/sec to prevent a reduction of the static oil pressure below the vapor pressure of the oil which would cause a cavitation condition. This criterion is based on previous successful operating ranges for Pratt & Whitney Aircraft oil pumps. It was necessary to reduce the gear diameter and change the number of teeth and gear pitch, compared to conventional pumps, to provide the required capacity without exceeding the tip speed limit for a 10,000 rpm speed operating condition.

d. Design Approach

(1) Utilization of Existing Hardware

The existing F100-PW-100 No. 2-3 bearing compartment test rig (No. F-34024) was chosen for testing the Compartmental Lubrication System critical components as an integrated system under engine conditions. This choice avoided the cost of an all-new rig. A primary design requirement of the Critical Component Design Task (Phase III, Task I) was to make the pump and tank as compatible with the existing rig as possible to minimize rig changes.

The system selected in Phase II of the study contract was based on an engine with the gearbox located on top of the engine to provide more volume for an integral tank at the bottom of the compartment. However, excessive costs required to modify to No. 2-3 compartment test rig for inverted operation dictated the use of a self-contained oil tank to keep oil off the high-speed pump drive gears driven off the normal towershaft location at the bottom of the compartment.

During Task I of Phase III, preliminary gear drive layouts were made to determine approximate gear diameters required to ratio the speed from 26,703 rpm at the towershaft gear to 10,000 rpm at the pump. These initial studies utilized the pump drive gear from an experimental

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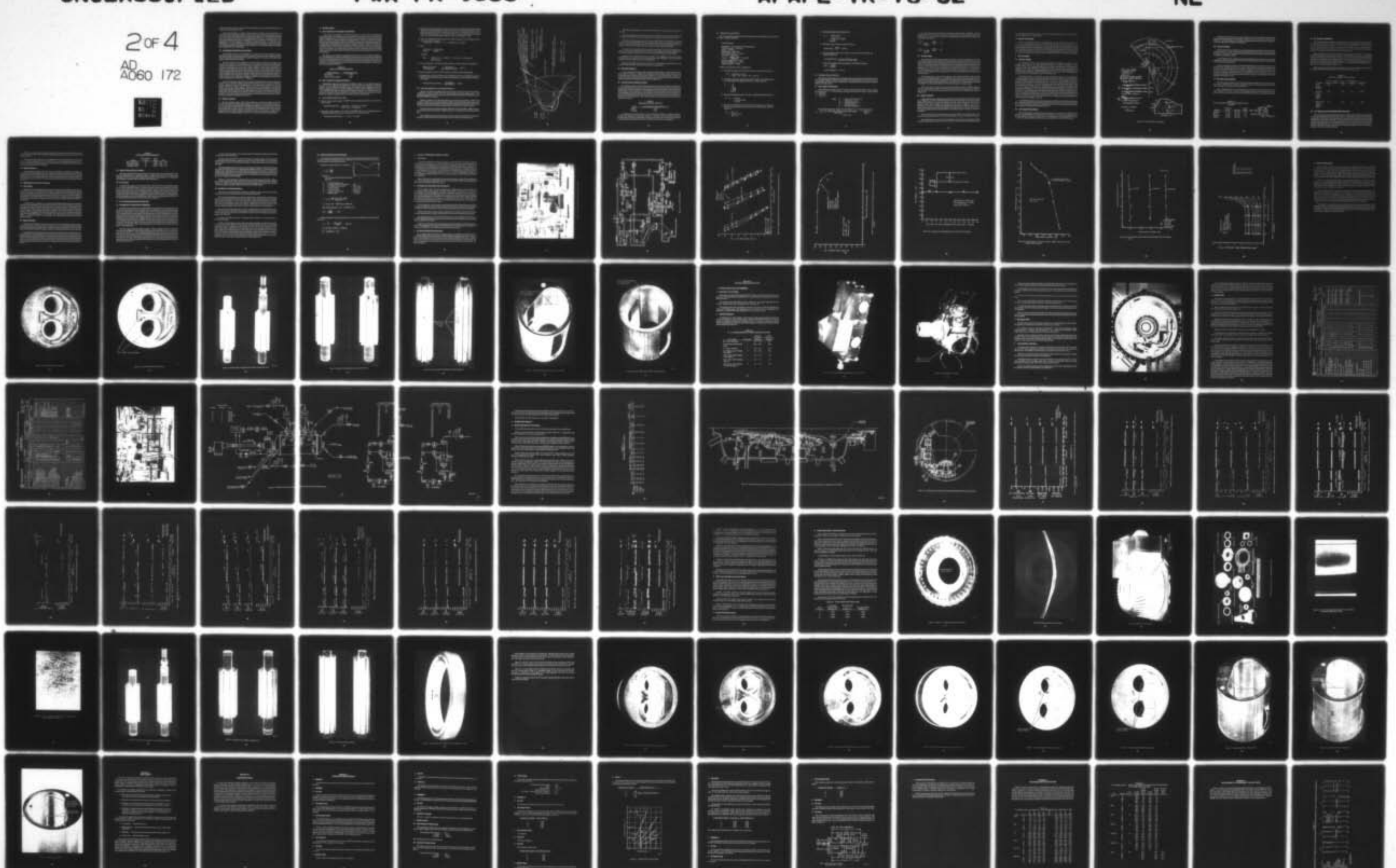


small engine (ST9) oil pump. New idler and towershaft gears were used. The pump location was based on these preliminary studies.

When Task III began, a review of existing Pratt & Whitney Aircraft gearboxes was conducted to find a double-gear idler shaft which could be meshed with new towershaft and pump gears to provide the proper speed reduction. A set of three gears used in the JT9D gearbox provide an oil pump speed of 10,158 rpm at a towershaft speed of 26,700 rpm. Although the hub of the pump and towershaft gears were not compatible with the pump shaft or towershaft gear, it was determined to be less expensive to reoperate existing gears than to make new gears. Subsequently, it was found that extra production gears were unavailable from the JT9D program, and it became necessary to fabricate new gears. However, the JT9D gear designs were utilized with modifications to the drive shafts.

(2) Compatibility With Existing Test Facilities

The full-scale system rig was designed to be tested in the Pratt & Whitney Aircraft, Government Products Division component test facility in D-area. Test stand modifications were limited to plumbing, instrumentation, and rig drive changes. The rig mount and coaxial drive gearbox existed from previous testing.

This test facility had the capability of setting bearing compartment conditions that simulate typical missions on advanced aircraft by controlling the air pressure and temperature around the compartment as well as the oil supply temperature and rig speed. Environmental conditions surrounding the compartment could be varied to match those corresponding to subsonic and supersonic flight points. Simulated altitudes from sea level to 60,000 feet can be run for the full range of speed conditions. The thrust loads on the No. 2 and 3 bearings were controlled by thrust balance pistons at the front and rear of the compartment. The stand can supply up to 3 lb/sec of airflow from subambient conditions to 200 psia at air temperatures from ambient to 1000°F. Oil flows can be varied and controlled up to 200 lb/min with temperatures from ambient to 400°F. Rig speed can be varied up to 14,000 rpm.

Control room instrumentation consisted of gages and manometers to monitor compartment pressures and air flowrates. Digital thermocouple temperature readouts allowed monitoring through multiposition switches. Temperatures of the compartments, bearings, and oil were closely monitored on digital readouts. Vibration levels of rig and gearbox were displayed continuously on meters. Digital readouts were used for monitoring oil flowrates, rig speed, and pump speed. Standard sharp edged, calibrated orifices were used for air flow measuring. Selected bearing outer race temperatures, rig internal vibrations, pressures, and speed were recorded on o-graph for continuous monitoring while on endurance running. Stand data were taken at regular intervals to provide for adequate data reduction.

(3) Design Constraints

The only constraint placed on the system design was to provide oil supply for an F100-PW-100 sized lubrication system while locating the oil supply and scavenge pumps, along with the oil tank, within an existing F100-PW-100 bearing compartment rig. This required running the pump at high speed to reduce the size of the drive gear train and pump volume. It also required reducing the tank volume by 8 percent, compared with the F100-PW-100 tank. Oil type used for sizing the pump was MIL-L-7808G. The deaeration system was required to deaerate up to 200 lb/hr of air to simulate leakage from the No. 1, 4, and 5 compartment labyrinth seals.

e. Oil Pump Design

(1) Gear Selection and Tip Speed Considerations

A gear pump was selected as the type of pump to be used for the Compartmental Lubrication System. Gear pumps are used for pressure and scavenge systems on most Pratt & Whitney Aircraft engines. Our experience with this type of pump allowed us to design with a high degree of confidence to ensure meeting the program objective. The gear size selected was based upon setting the tip speed about equal to our standard 7-tooth, 6-pitch straight spur pump gear. This gear runs at shaft speeds on other engines from 2500 to 4000 rpm. The speed selected for the Compartmental Lubrication System pump was 10,000 rpm and was based upon the desire to increase the speed to the maximum allowable and to reduce the size of the pump and drive gear train to fit into the No. 2-3 compartment rig. Experience with a 10,000 rpm pump on the UTTAS engine demonstrator (ST9) program indicated we could meet the 50-hour endurance test set forth in the contract.

The final gear size selected for the high speed pump was a 9-tooth, 16-pitch straight spur gear. The displacement of this gear (0.1686 in.^{3}/in. of face width) gave a reasonable face width for the capacity required and kept the tip speed approximately equal to experience levels. Gear tooth loading was an insignificant factor in selecting this size gear because the gear tooth stresses are extremely low. Calculations for the gear tooth stresses are presented in Appendix K. Design safety factors are presented in Table 20.

TABLE 20
GEAR TOOTH DESIGN MARGIN

Design Parameter	*Design Safety Factor*
Hertz Stress	2.038
Dynamic Tooth Loading	1.405

(2) Gear Length and Leakage Calculations

Required gear length was calculated by two different methods: (1) by scaling the gear teeth 50 times size and measuring the displacement between teeth, then applying a volumetric efficiency; (2) by scaling the measured output and calculated effective leakage area of a low capacity experimental pump of a similar configuration (ST9 pump for UTTAS demonstration) up to F100-PW-100 output flow requirements. Excellent agreement was obtained by the two methods of calculation. These calculations were confirmed by measured pump capacity values during the component and system tests. An outline of the procedures follows:

(a) Pump Capacity Scaled From Layout

- Required supply pump capacity is F100-PW-100 intermediate power flow plus a 15 percent over capacity.

Required Supply Flow = 152.5 ppm + 15 percent over capacity
= 152.5 + 22.9 = 175.4 lb/min

- Required scavenge capacity is two times the F100-PW-100 No. 2-3 compartment flow at intermediate power to allow for an air-oil mixture (two component flow).

Required Scavenge Capacity = 2 × 88.6 = 177.2 ppm.

- Figure 22 shows the mesh between the two 9-tooth, 16-pitch, 28-degree pressure angle straight spur gears for the high speed pump, scaled 50 times size. The cross-hatched area is the pump displacement between teeth. Pumping occurs when the oil is displaced between the pump gear teeth and the housing sleeve on opposite sides of the pump, 180 degrees from the gear mesh. The calculated area between teeth was found to be 0.009367 in.^{2}/tooth.

- Under the conditions of 9-teeth per gear and two gears pumping, the pump displacement per inch of gear length is given by:

  Displacement = 0.009367 × 9 × 2 = 0.1686 in^2/rev-in. of length.

- Given:

  Gear Speed = 10,000 rev/min
  Density = 60 lb/ft^3

  Therefore:

  Flow = 0.1686 in^2/rev-in. × 60 lb/ft^3 × 1 ft^3/1728 in^3 × 10,000 rev/min
  = 58.54 lb/min-in.

- For a required flow of 175.4 lb/min and an assumed volumetric efficiency of 88 percent:

$$\text{Pressure Pump Face Width Required} = \frac{175.4 \text{ lb/min}}{(58.54 \text{ lb/min-in.})(0.88)} = 3.4100 \text{ in.}$$

This gear width had to be split in half to provide acceptable journal bearing lengths.

- Scavenge pump volumetric efficiency was considered to be close to 100 percent due to low pressure rise across this pump and consequent low leakages. At a required flow of (2)(88.6) = 177.2 lb/min:

$$\text{Scavenge Pump Face Width} = \frac{177.2 \text{ lb/min}}{58.54 \text{ lb/min-in.}} = 3.027 \text{ in.}$$

(b) Pump Size Scaled From Low Capacity ST9 Pump

Because test data was available from the ST9 pump which had a nonconventional gear configuration (9-tooth, 16-pitch), like the Compartmental Lubrication System pump, it was decided that a good check on the pump size could be obtained by scaling the ST9 pump up to the required F100-PW-100 oil flows.

Running clearances were calculated taking into account thermal growths at 300°F using minimum, maximum, and nominal dimensions. Where actual ST9 pump measurements were available; however, these values were used to calculate leakage areas because they corresponded closely with the maximum tolerances and could also be correlated with the pump data.

ST9 oil flow data was corrected for density differences between the MIL-L-23699 oil used for the small pump tests and the MIL-L-7808 oil used for the Compartmental Lubrication System tests.

Three leakage paths were identified for the pump: (1) past the end plates, (2) through the clearance between the gear teeth and the liner, and (3) between the housing and liner.

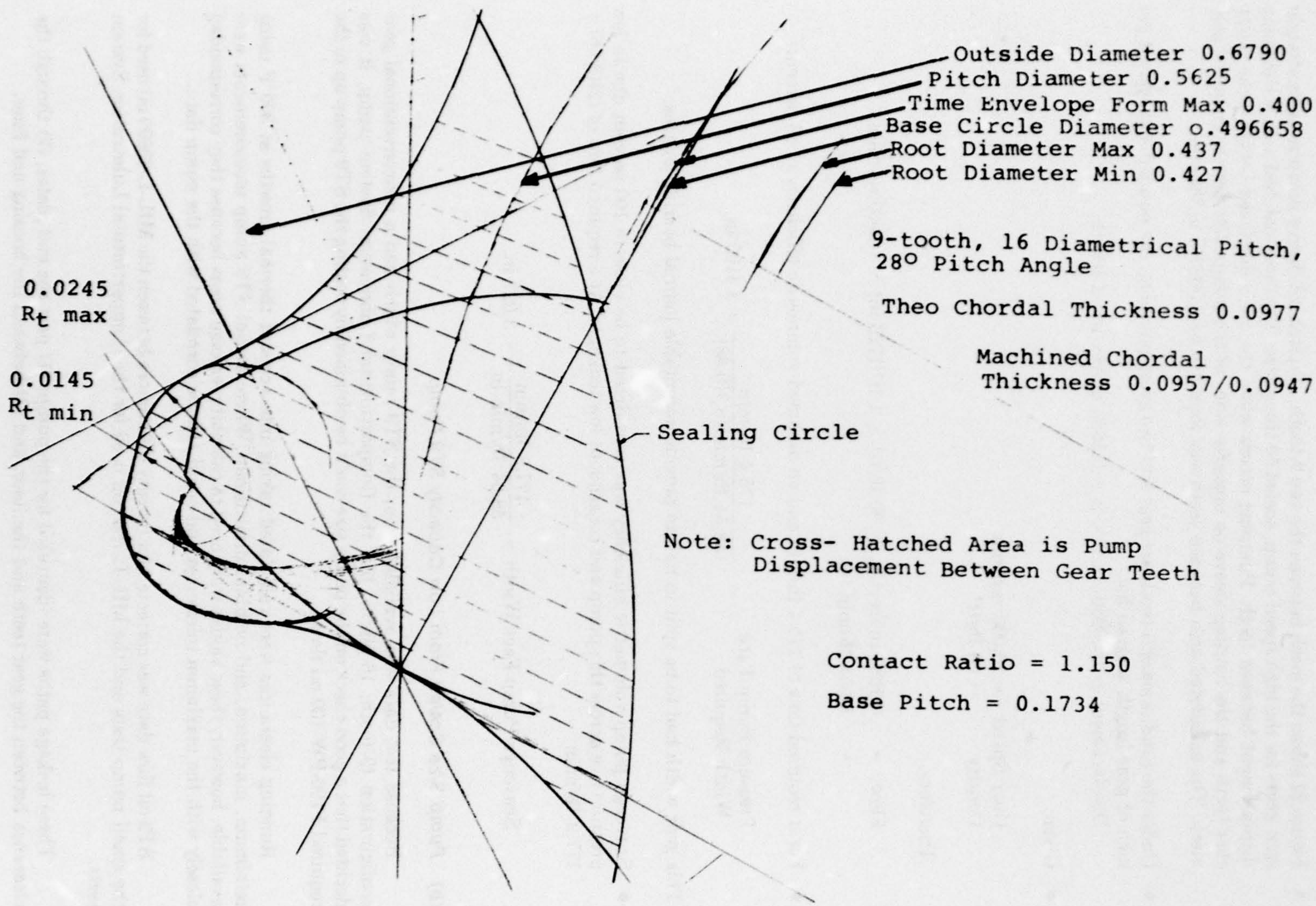


Figure 22. Compartmental Lubrication System Pump Gears 50 Times Size

The ST9 pump leakage was calculated as the difference in output flow at 0 psid and at 150 psid.

End plate leakage for the new pump was assumed to be the same as that of the ST9 pump while gear tooth and housing-to-liner leakages were evaluated as a function of gear length.

Pressure loss constants were calculated for each leakage path. The end plates were treated as orifices. The gear tooth leakage path was treated as three orifices in series because three teeth can be in contact with the liner at a given time. The housing-to-liner flow path was treated as an inlet and exit loss plus a frictional loss, plus a loss due to the leakage path length.

An equation was formulated with the required output flow equal to the no-leakage pump capacity as a function of length minus the shell-to-housing leakage as a function of length minus the end plate leakage. This equation was then solved for required pump element length.

Leakage for the scavenge pump was assumed to be negligible because the pressure differential across the pump is small. Pump length was then calculated as the required flow divided by the flow capacity per inch of pump element.

Pump element length was calculated to be 3.35 in. for the supply pump and 3.03 in. for the scavenge pump. Detailed calculations are presented in Appendix K.

(3) Shaft Seal Selection

It was determined in the early stages of the pump design effort to use seals on the pressure pump shafts to eliminate a leakage path through the shaft journals. This proved to be an economical means to improve the volumetric efficiency on the pressure pump. A teflon type spring loaded radial shaft seal marketed by the Flurocarbon Company, Mechanical Seal Division under the trade name Tec-Ring, was selected for this application.

(4) Journal Bearing Loading and Sizing

The approach used in sizing the high-speed bearing journals was to design to the same unit loading as the F100-PW-100 pump journals while maintaining other design criteria of minimum Sommerfeld Number and maximum journal length-to-diameter ratio based on Pratt & Whitney Aircraft engine oil pump experience. The F100-PW-100 uses carbon insert journals similar to the Compartmental Lubrication System pump. Because the F100-PW-100 pump journals have a design life of 6000 hours and have been trouble-free in production engines, it was concluded that this approach would provide for a safe design. Supply and scavenge pump journal lengths are presented in Table 21.

TABLE 21
REQUIRED JOURNAL LENGTH

Pump	*Journal Length Per Element End ~ in.*
Supply	0.491
Scavenge	0.250

A procedure has been developed at Pratt & Whitney Aircraft for calculating resultant bearing loads taking into account the variation in pressure around the pump. Using this procedure, the unit load on the F100-PW-100 journals was found to be 443.9 lb/in^2 The derivation of this analysis and the detail calculations of these results are shown in Appendix K.

(a) Pressure Pump Journal Size

The Compartmental Lubrication System pressure pump journal length for a unit load of 443.9 lb/in.[2] was then calculated:

Pump Parameters

Horsepower = 2.0 per each of two pressure pumps
Nominal Flowrate = 150 lb/min
Density = 59 lb/ft^3
Pump Speed = 10,000 rpm
Gear Face Width (W_r) = 1.675 in.
Pump Rise = 150 lb/in^2

$$\text{Torque} = \frac{63{,}000 \times 2\ \text{HP}}{10{,}000\ \text{rpm}} = 12.6\ \text{in.-lb}$$

Gear Pitch Radius (R) = 0.281 in.
Gear Outer Radius (r) = 0.340 in.
Pressure Angle (θ) = 28 deg.

Based on the derivations given in Appendix K:

- The hydraulic load in the X-direction (toward the pump inlet) is given by:

$$\begin{aligned} F_{HX} &= 1.636\ (W_r)\ (\gamma)\ (P_{max}) \\ &= 1.636 \times 1.675 \times 0.340 \times 150 = 139.75\ \text{lb} \end{aligned}$$

- One-half of the torque is transmitted to the driven gear and one-half absorbed by the driver gear. The tangential load due to torque is given by:

$$\begin{aligned} F_t &= \tfrac{1}{2}\,\frac{T}{R} \\ &= \tfrac{1}{2}\,\frac{12.6}{0.281} \\ &= 22.42\ \text{lb} \end{aligned}$$

- The gear teeth separating load is the only y component load and is given by:

$$\begin{aligned} F_y = F_s &= F_t \tan\theta \\ &= 22.42 \tan 28\ \text{deg} \\ &= 11.92\ \text{lb} \end{aligned}$$

- The idler gear absorbs the major load because the hydraulic and tangential loads are in the same direction. The resultant X component load on the idler is given by:

$$\begin{aligned} F_{IX} &= F_{HX} + F_t \\ &= 139.75 + 22.42 \\ &= 162.17\ \text{lb} \end{aligned}$$

- **The resultant idler load is then given by:**

$$F_I = \sqrt{F_{Ix}^2 + F_{Iy}^2}$$
$$= \sqrt{(162.17)^2 + (11.92)^2}$$
$$= 162.60 \text{ lb}$$

- **The load on each of the two journals is given by:**

$$\text{Journal Load} = \frac{162.60}{2} = 81.30 \text{ lb}$$

- **The unit pressure load on the journal is the journal load divided by the projected journal area:**

$$\text{unit pressure load} = \frac{F_I}{\text{(journal dia) (journal length)}}$$

Using the unit pressure load calculated for the F100-PW-100 pump:

$$443.9 = \frac{81.30}{(0.373)(L)}$$

$\therefore$ **Journal Length (L) = 0.491 in.**

(b) Scavenge Pump Journal Size

The required journal length for the scavenge pump was calculated using the same procedure as for the pressure pump. As shown in Appendix K the required length was only 0.079 inches. A journal length of 0.250 inches was selected based on experience from Pratt & Whitney Aircraft designed scavenge pumps.

(c) Other Design Considerations

Pratt & Whitney Aircraft practice is to provide a minimum Sommerfeld No. for an oil pump journal bearing of 5×10^{-4}. This is based on studies of JT3 and JT8 oil pumps. The Sommerfeld No. is defined as:

$$S = \frac{\mu N}{P}\left(\frac{R}{C}\right)^2$$

where:
- μ = viscosity of oil, $\text{lb}_f\text{-sec/in}^2$
- N = shaft speed, rev/sec
- P = Projected pressure, psi
- R = Journal radius, in.
- C = Diametral clearance, in.

For the high speed pump journals, the Sommerfeld No. is well above this criteria.

$$S = \frac{(2.91 \times 10^{-7} \text{ lb}_f \text{ sec/in}^2)\ (10{,}000 \text{ rev/min})\ (1/60 \text{ min/sec})}{443.9 \text{ lb/in}^2} \left(\frac{0.1865 \text{ in.}}{0.0015}\right)^2$$
$$= 16.89 \times 10^{-4}$$

It is also design practice, based on a number of pump designs, to maintain a journal length/diameter of less than 1.50. If a bearing is excessively long, the bending of the shaft in the journal may cause journal-bearing contact at the bearing ends. As shown below, the high speed pump meets this criteria.

$$\left(\frac{L}{D}\right)_{\substack{\text{supply}\\ \text{journals}}} = \frac{0.491}{0.373} = 1.32$$

$$\left(\frac{L}{D}\right)_{\substack{\text{scavenge}\\ \text{journals}}} = \frac{0.250}{0.373} = 0.67$$

(5) Housing Design

The pressure pump and scavenge pump were stacked in series and packaged into a single pump housing assembly. One element of the pressure pump was driven directly off a drive gear, and the other element and the scavenge pump was driven through quill shafts from the first pump element. The housing assembly was made up of a center housing for the gears and bearings and a pressure relief valve plus two side housings which are manifolds for the oil in and oil out. Because only two sets of pump housings were purchased for this project, it was decided to machine the housings from plate stock rather than to design and purchase cast housings. A cast housing could be designed to reduce weight and size as well as complexity but was not warranted for this program.

The housing stress is very low. The maximum stress within the housing is the flat plate stress on the discharge manifold due to 150 psid ΔP across the wall. The stress margin of safety at this location is 5.94 as shown in Appendix K. The pump housing mount lugs that attach to a ring in the rig are lightly loaded which results in minimal stresses. The mount lugs were designed for stiffness to ensure proper alignment of the pump drive gear to the drive mesh.

The criteria used for sizing the inlet and exit manifolds are based on P&WA experience factors to ensure smooth steady oil flow within the pump system. The inlet line was sized for a flow of 5 ft/sec while the exit line was sized for a flow of 15 ft/sec. Calculations are shown in Appendix K.

(6) Material Selection

A high durability pump was the prime consideration in selecting material for the high-speed pump. The gears are made of AMS 6470 and the gear teeth and shafts are nitrided to a DPH hardness 850 minimum to ensure good surface wear. The journal bearings are constructed of graphitic carbon sleeves pressed into housings. Press-fit stress calculations are shown in Appendix K. This type of carbon bushing has been demonstrated in P&WA engines to be a simple, durable-type journal capable of long life in oil supply and scavenge pump environments. No special pressure grooves are required within the bearing journal to maintain an oil film for lubrication.

The pump housing and bearing housing are made of AMS 4117 aluminum alloy. This material was selected primarily for the ease of machinability and good strength characteristics.

The quill shafts used to transmit torque through the pump stages and to the scavenge pump are made of AMS 6488 tool steel. The teeth are nitrided to a case hardness of DPH 850 minimum

to ensure long wear life. Torque capacity as well as tooth bearing and shear stress calculations for the quill shaft are given in Appendix K.

(7) Bypass Valve Design

Because the oil flow and pressure requirements for the high-speed pump are identical to those of the F100-PW-100 engine, the F100-PW-100 bypass valve was selected for the Compartmental Lubrication System pump. The internal components of the F100-PW-100 valve were used and fitted into the high-speed pump housing. The valve is a spring-loaded, movable seat configuration that seals on a fixed valve. The valve assembly is adjusted to bypass oil from the pump discharge back to the pump inlet at pump pressure differentials above 175 psid. This is to protect the housing from excessive pressure during cold oil starts or downstream blockage.

f. Oil Tank Design

(1) Deaerator Design

A comprehensive test program was conducted at Pratt & Whitney Aircraft in 1974 to evaluate twelve different oil tank deaeration schemes for possible incorporation in the F100-PW-100 oil tank. A windowed tank was used to allow visual observation of the deaeration phenomena. The visual differences in deaeration qualities among the various schemes were so apparent that no analytical evaluation method was required. The selected scheme has been included as Bill-of-Material on the F100-PW-100 engine since November 1975 as P/N 4044275 and was selected for the Compartmental Lubrication System tank.

The deaerator body is a cylindrical can type configuration. The air-oil mixture enters the cylinder tangentially at the top and at an angle with the cylinder centerline so as to centrifuge the oil around the ID of the deaerator and direct it toward the bottom to prevent splash out at the top. The bottom of the cylinder is covered, and four slots ½ in. by 2 in. are provided on the side of the deaerator, near the bottom to allow the deaerated oil to flow to the bottom of the tank. The air exits through a curved pipe at the top of the deaerator. The 1974 testing also disclosed that ⅛-in. holes, drilled in the entry tube, discharged mostly air and reduced the violence of the discharge into the deaerator. Ten holes, similar to the F100-PW-100 design, are included in the entry tube to the Compartmental Lubrication System deaerator. The air/oil mixture enters a circular-to-rectangular transition as it is fed into the deaerator to flatten the flow and provide for a better distribution of the flow within the cylinder. The deaerator is mounted at the same level relative to the full oil level as in the F100-PW-100 but at a slightly more inclined position from vertical because of the shape of the tank.

The F100-PW-100 deaerator separates approximately 60 lb/hr of air while the rig deaerator had to separate 200 lb/hr of air due to the labyrinth seals used in place of the carbon seals in the No. 1, 4, and 5 bearing compartments. Component bench tests have demonstrated the capability of this deaerator to handle the required air/oil flows.

(2) Tank Capacity Calculations

The oil tank capacity, calculated as shown on Figure 23, was found to be 2.78 gallons. The full level was determined by the placement of the deaerator in the tank and the requirement to have the full level at the same location relative to the deaerator as in the F100-PW-100 oil tank in order to maintain deaeration conditions as close as possible to the F100-PW-100 tank.

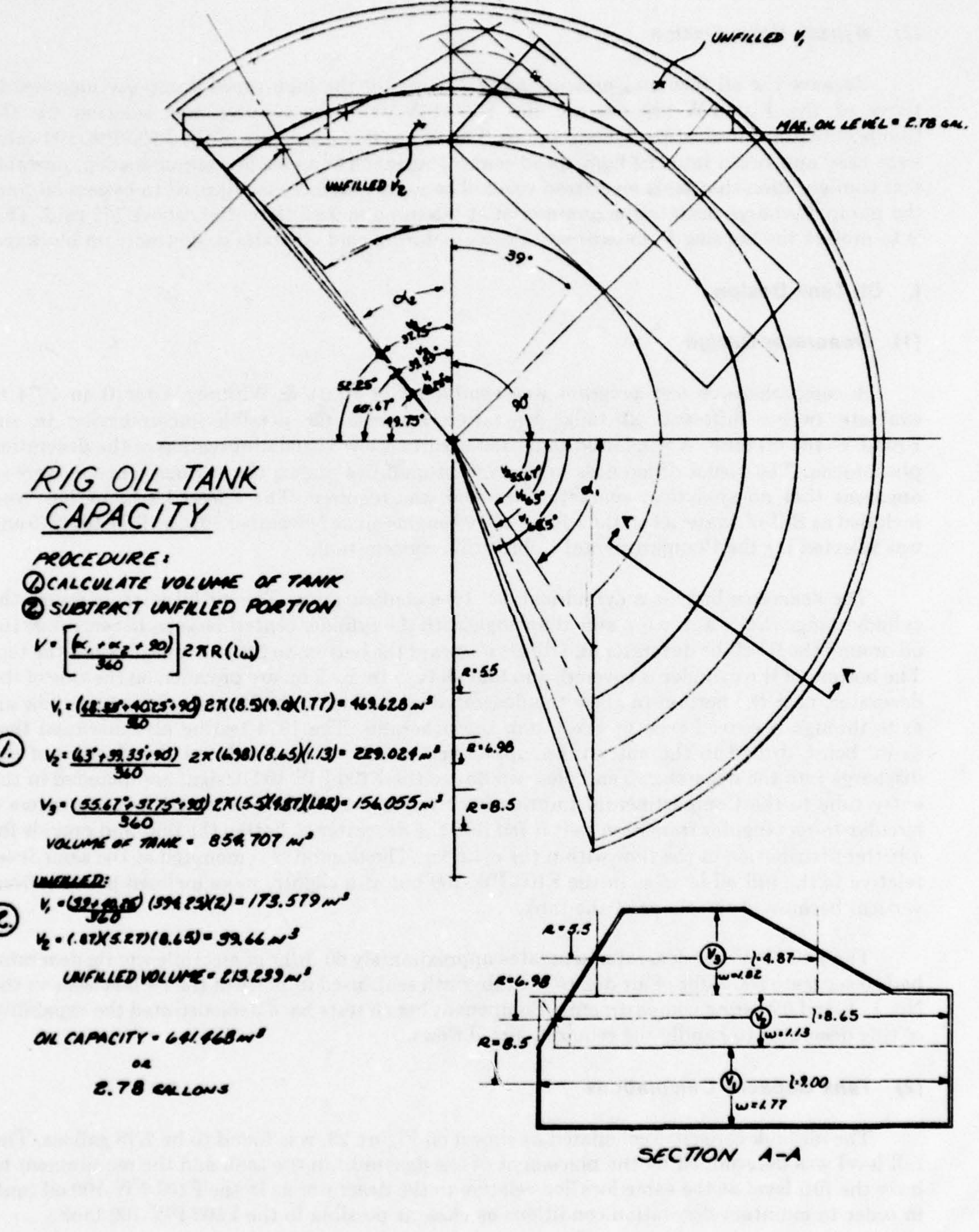


Figure 23. Oil Tank Volume Calculations

A dipstick was provided with the tank to measure oil levels. This dipstick was calibrated during the component bench tests by adding oil, one quart at a time, and marked using the noted location of the wetted indication on the dipstick. This procedure revealed the actual 2.75 gal level to be 0.4 in. above the calculated level of Figure 23.

(3) Internal Plumbing

Oil from the scavenge pump flows into a tee inside the oil tank, through a short jumper tube, and mixes with No. 1, 4, and 5 compartment oil entering another leg of the tee. The oil then flows through a 7-in. long tube to the deaerator inlet. This is the tube with holes mentioned previously in describing the deaerator design.

A 1-inch diameter tube is welded from the outer wall to the inner wall of the tank to provide a passageway for an oil-in tube which supplies oil to the engine bearings. Another 1-inch diameter tube is the guide for the dipstick.

A boss is provided at the bottom of the tank for draining the oil, and a port at the top of the tank allows the air separated from the oil to escape. This top port also allows the pressure in the tank to be reduced with a stand mounted ejector to simulate various altitude flight conditions for the system tests and to check the pump's suction capabilities during bench tests. All external fittings were provided with threaded holes to attach fittings for the component bench tests. During the system test, the fittings were attached to the rig outer case and plugged into the tank plumbing with piloted O-rings.

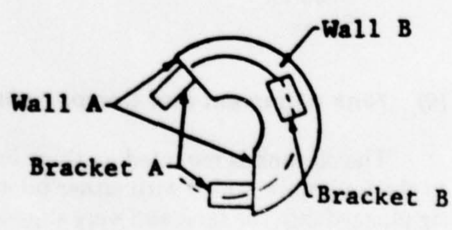
(4) Mounting Flange Analysis

The oil tank was supported at three locations by brackets welded to the outer surfaces of the tank. Stress calculations are included in Appendix L and are summarized in Table 22. Stresses were calculated for a 10g load in the axial direction.

Each of the brackets are fully supported by either a mount ring which is part of the rig outer case (Flange A) or by the No. 2 bearing and seal support flange. This reinforcement limits the deflection and hence the stress in the mount brackets and the walls to which they are welded.

TABLE 22
OIL TANK MOUNTING FLANGE STRESS CALCULATIONS

Region	*Calculated Stress*	*Allowable*	*Safety Factor*
Bracket A	12 ksi	94.5 ksi	7.9
Wall A	7.4 ksi	94.5 ksi	12.8
Bracket B	8.5 ksi	94.5 ksi	11.1
Wall B	72.7 ksi	94.5 ksi	1.3



(5) Tank Pressure Capabilities

The tank experienced no pressure differential during system testing because the breather port was open to the rest of the rig which surrounds the tank. During component testing, however, the tank internal pressure was reduced to 2 psia or a pressure differential of 12.7 psid. The large outer surface of the tank was subject to external ambient pressure and, thus, the possibility of buckling existed. In addition, the large conical surfaces at the front and rear inner surfaces were subjected to large loads resulting in significant bending stresses.

Stress calculations are shown in Appendix L and summarized in Table 23. A pressure differential of 12 psid was utilized in the calculations. Minimum factors of safety of 4 for buckling and 3 for bending stress were established. Because of the welded box structure of the tank, all surfaces were considered to be complete rings. A tank wall thickness of 0.031 in. was first considered, but this resulted in a buckling pressure of 16.9 psid and an unsatisfactory factor of safety of 1.4. The selected wall thickness of 0.062 in. provided a critical buckling pressure of 95.4 psid and 2 times the required safety factor.

Depending on the method of support considered for the two conical surfaces, the stress could be as high as 20,040 psi for the forward cone and 6,400 psi for the rear cone.

TABLE 23
OIL TANK STRESS CALCULATIONS

Location	*Critical Pressure*	*Buckling PSID*	*Factor of Safety*	*Required Factor of Safety*	*Stress PSI*
Outer Surface Buckling	95.4		8.0	4.0	NA
Forward Conical Surface Bending Stress	NA		3.94	3.0	20,040
Rear Conical Surface Bending Stress	NA		12.34	3.0	6,400

(6) Tank Alignment and Compatibility With System Rig

The oil tank is mounted on three brackets as previously described. All fittings which attach to the tank were sealed with either piloted O-rings or conical gaskets. Fittings from outside the rig plugged into the tank and were supported by the rig outer case. Clearance between the fittings and the holes in the rig case is sufficient to accommodate the location tolerance between the tank and the rig case. The large 1.25-in. diameter tube from the tank to the pump inlet was the stiffest plumbing component and had to be installed before the tank was secured in final position. A stress of 17,655 psi in the tube would result from imposing a deflection equal to the tolerance stackup on the tube's installed endpoints.

All bosses on the tank were designed for use during system testing as outlined above and had threaded holes provided to allow attaching external plumbing during component testing of the pump and tank.

The tank external configuration was established by the internal configuration of the existing No. 2-3 bearing compartment test rig and the three gallon capacity requirement. Due to limitations imposed by the rig geometry and the placement of the deaerator, the resulting tank capacity was calculated to be 2.78 gallons.

(7) Material Selection

The oil tank was made from AISI 410 stainless steel. While this material is less resistant to corrosion than AISI 300 series stainless steel, its greater strength was required due to the buckling loads imposed by the reduced pump inlet pressure testing performed during the component bench tests. In an actual engine application, this buckling problem would not exist because the tank breather is open into the No. 2-3 bearing compartment, and the pressure differential would not exist.

g. High-Speed Drive Train for Oil Pumps

(1) Gear Design

The gears used on the system test rig were based on gears from the JT9D main gearbox. The original intent was to use the JT9D idler shaft without modification and modify the two adjacent gears to fit the shafts in the system test rig. This procedure would have eliminated the need to have special gears cut and would have required only new hubs to be cut on two gears. Production requirements of the JT9D program were such, however, that no existing gears were available for this program. Consequently, the rig gears, based on JT9D gear designs, had to be procured by a special order.

The gear teeth were checked in accordance with the P&WA design procedures for spur gears and found to be adequate for the loads transmitted in the system test rig. Calculations are shown in Appendix M. Pratt & Whitney Aircraft Computer Program No. 5905 for calculating spur gear tooth thickness reduction was run with the gear-to-gear tolerances from the system test rig input into the program. This program calculated a required tooth thickness reduction of 0.004 to 0.008 in. for the towershaft-to-idler gear mesh and 0.007 to 0.011 in. for the idler-to-pump gear mesh. The actual JT9D gears are manufactured with tooth thickness reductions of 0.0055 to 0.0095 in. and 0.0075 to 0.0115 in., respectively. The JT9D gears thus meet structural and geometry criteria.

(2) Bearing Selection

Existing Pratt & Whitney Aircraft parts were selected for all bearing locations in the oil pump drive gear train. The idler gear was supported on two conrad ball bearings. These bearings were axially loaded with a spring washer to positively locate the gear shaft. This also provided proper thrust load for satisfactory operation with the very light radial load due to gear reactions.

The towershaft bevel pinion gear shaft in the rig was supported by two ball bearings. In an engine application, the towershaft pinion would be supported by one ball bearing and one roller bearing because of the much higher gear reaction loads resulting from the high gearbox power extraction. This rig application results in such low gear reaction loads that an internal load from a spring washer was required to provide sufficient thrust load for satisfactory operation of the ball bearings. Spring washer calculations are given in Appendix M. Bearing parameters are shown on Table 24.

TABLE 24
PUMP DRIVE TRAIN BEARINGS

Position	*Bore Diameter (mm)*	*Speed (rpm)*	*DN*
Idler (2 locations)	20	17,307	0.35×10^6
Towershaft (upper)	35	26,703	0.935×10^6
Towershaft (lower)	50	26,703	1.335×10^6

(3) Support Flange Structural Analysis

The loads transmitted from the gear train to the support structure are extremely low. The structural analysis given in Appendix M shows a design safety factor of 16.9. The design philosophy was to fit the gear train support system rigidly in the existing No. 2-3 compartment rig and position the gears closely for smooth power transmission.

(4) Oil Jet Sizes

A manifold was tapped off the oil supply line to provide lubrication for the idler bearings and the lower towershaft bearing. The upper towershaft bearing was lubricated by existing No. 2-3 compartment oil jets. The oil jets were sized as shown in Appendix N. Taking the full pressure loss across single jets at the bearings would have resulted in very small jets which could be easily blocked by contamination. It is practice at Pratt & Whitney Aircraft to limit minimum oil jet sizes to 0.035 in. This was accomplished by providing a flow restriction at the manifold inlet to reduce the pressure at the individual oil jets. The resulting oil jet diameters were 0.058 in. at the manifold inlet and 0.048 in. at each of the bearing supply lines.

h. No. 2-3 Bearing Compartment System Rig

(1) Arrangement of Components and Alignment

Looking aft, the oil tank was on the right half of the compartment, the oil pump was on the left side, and the gear train was at the bottom of the compartment. The tank and main supply pump were connected by a 1.25-in. diameter tube, and the scavenge pump was fed by another 1.25-in. diameter tube which extended to the bottom sump area of the compartment. The tubes both curved to the left to avoid interference with the new front support for the No. 2 bearing. The discharge from the scavenge pump passed through a short jumper tube which was trapped between the pump and the tank by a shoulder and snap ring on the jumper tube. The jumper tube was inserted into the tank as far as possible by moving the snap ring as far onto the jumper tube as possible. The tank was installed in the rig, the jumper tube inserted into the hole in the pump housing until the shoulder contacts the pump housing, and the snap ring was installed into the groove in the jumper tube.

The main pump discharge passed through a fitting 15 degrees below the horizontal centerline which plugged into a hole in the pump housing and was bolted to a flat on the rig outer case. Oil from the No. 1, 4, and 5 bearing compartment simulator joined the scavenge pump discharge oil in an internal tee in the oil tank after passing through a fitting which was also bolted to the rig outer case and was piloted into a hole on the tank. The same type fitting was used at the bottom of the rig as a drain plug for the tank and rig. A cap closed the fitting during rig operation. Removal of the cap drained the tank, and removal of the fitting drained both the tank and sump area of the rig.

A cover on the outer surface of the rig sealed a port through which the dipstick was inserted to check the oil level in the tank.

The gear train was bolted to the bottom of the No. 2 bearing support where the bottom towershaft gear bearing support normally is located. The plate was located by a pilot diameter and a dowel pin to locate the idler gear shaft angularly to ensure proper gear mesh with the pump drive gear.

The idler gearshaft and the lower bevel gear bearing were attached to a large flat plate which was bolted to the bottom of the No. 2 bearing support (Reference Figure 17). The oil pumps were located by two dowel pins in the rig outer case. The pumps were mounted to a portion of an existing flange within the No. 2-3 rig. The remainder of the flange was cut away to provide room for the oil tank and the gear train components. An oil distribution manifold wrapped around the gear train to supply oil to the bearings.

Tolerance on the towershaft-to-idler mesh and on the idler-to-pump mesh was ± 0.009 in. and ± 0.016 in. respectively. These tolerances were used in Pratt & Whitney Aircraft Computer Program No. 5905 to calculate required tooth thickness reduction. The tolerances are less than could be tolerated by the gear meshes providing for an acceptable design.

(2) Modification to Existing Hardware

The outer case of the existing rig, F34024, was modified to make room for the tank and pump and to provide mount provisions for external fluid connections.

The forward internal flange was cut almost entirely away to make room for the tank, leaving lugs for mounting the tank and pump. Flats were added to the outer surface for external fittings. These fittings included No. 1, 4, and 5 scavenge return, main oil pump discharge, tank drain, dipstick port, thrust piston air inlet, and a cover for a pump drive gear clearance slot. All flats were at the same dimension from the rig centerline. A sheet metal cover was welded over the normal towershaft opening at the bottom of the case to keep the oil in the compartment.

Large clearance cutouts were made in the No. 2 bearing support to clear the pump housing, the idler gears, and the upper idler bearing support. External ribs and bosses were removed to clear the tank and pump. A dowel pin was added at the lower rear surface to align the plate which supports the idler shaft.

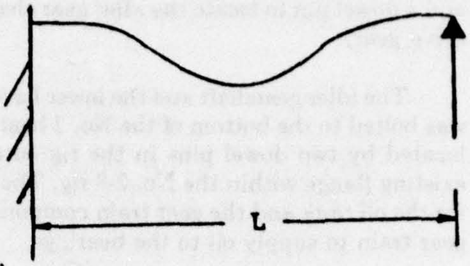
A fitting was welded to the No. 2 bearing nozzle to supply oil to the gear train oil distribution manifold.

In order to obtain sufficient volume in the rig oil tank, the engine type forward support had to be eliminated. This would not be necessary in an engine application of the Compartmental Lubrication System configuration since making the walls of the tank integral with the compartment walls would provide the required tank volume. A new support was required which had to duplicate the radial spring rate of the engine part. The Yale Shell Analysis Computer Program No. 8330 was utilized with a saddle load applied to calculate the radial spring rate of the new part. A simple cone and cylinder arrangement yielded a radial spring rate of 1.7×10^6 in./in. compared with 1.5×10^6 in./in. for the engine part. The stiff cone provides a stable mount for the carbon face seal, and the thin cylinder provides the desired spring rate.

(3) Internal Plumbing Structural Analysis

The rig internal plumbing was short in length and stiff resulting in high natural frequencies above any existing driving frequencies. For example, the oil in line was the worst case and its natural frequency was calculated as follows:

Assumed a pinned, fixed beam 17 inches long



$$f_1 = K \sqrt{\frac{gEI}{WL^4}}$$

Reference:

Kent Mechanical Engineering Handbook, page 9-03

f_1	=	natural frequency, cps		
K	=	constant for pinned, fixed train	=	2.45
g	=	acceleration due to gravity	=	386 in./sec^2
E	=	Youngs modulus	=	30×10^6 psi
I	=	moment of inertia	=	0.0247 in.^{4}
W	=	weight of beam per inch	=	0.040 lb/in.
L	=	length of tube	=	17 inches

$$f_1 = 2.45 \sqrt{\frac{386 \times 30 \times 10^6 \times .0247}{(0.04)\ (17)^4}}$$

$$f_1 = 716 \text{ cps} \times 60 = 43{,}000 \text{ cpm or } 43{,}000 \text{ rpm}$$

Max exciting frequency in rig = 13,900 rpm rotor

$$SF = \frac{43{,}000}{13{,}900} = 3.09$$

The hoop stress due to the internal pressure is very low; for example the oil out line pressure stress was:

$$S = \frac{Pr}{t} = \frac{150 \times .375}{0.035} = 1607 \text{ psi}$$

2% yield PWA 770 Mat'l = 20,000 psi

SF = 20,000/1607 = 12.44

2. CRITICAL COMPONENT CHECKOUT TESTS

a. Test Set-Up

The pump and tank were mounted in D-area, D-4 stand, as shown in Figure 24. The pump was mounted to, and driven by, a 15 hp Varidrive DC motor. The tank was mounted such that the distance from the pump inlet to the oil level in the tank was the same as in the No. 2-3 compartment rig. A breather tank was mounted directly above the main oil tank. Aeration of No. 2-3 compartment oil was achieved by injecting air into the tank shown to the right of the main oil tank. Fittings for oil in, oil out, air in, and air out and instrumentation provided a model of the engine No. 2-3 compartment. Aeration of No's. 1, 4 and 5 compartment oil flows was accomplished by injecting air directly into the oil line returning to the tank. Figure 25 shows the stand schematic.

Figure 18 shows the compartmental oil tank used for these tests. Figure 19 is a disassembled view of the pump housing, gearshafts, and sleeves. An F100-PW-100 Bill-of-Material pressure relief valve (used in this pump) is also shown.

b. Oil Supply and Scavenge Pump Performance

The oil supply pump was run at speeds up to 10,000 rpm (two and one-half times conventional engine pump speeds) delivering F100-PW-100 oil flowrates. Sixty hours of accumulated run time was logged on two pump assemblies (20 hours on S/N 1, 40 hours on S/N 2) without any performance deterioration. A pump map was generated from the observed test data for each assembly at 7000, 8500, and 10,000 rpm pump speeds. These maps, illustrating the delivered oil flowrate-versus-pressure rise characteristic, are shown in Figure 26. The Equipment Test Plan guarantee flowrate (superimposed on pump map) was met satisfying the contractual goal for delivered flow output.

The oil supply pump inlet pressure was reduced from ambient to approximately 2 psia while operating at 10,000 rpm. At the guarantee point (4 psia inlet pressure) insignificant flow fall-off was observed. This is illustrated in Figure 27 with the guarantee point shown superimposed.

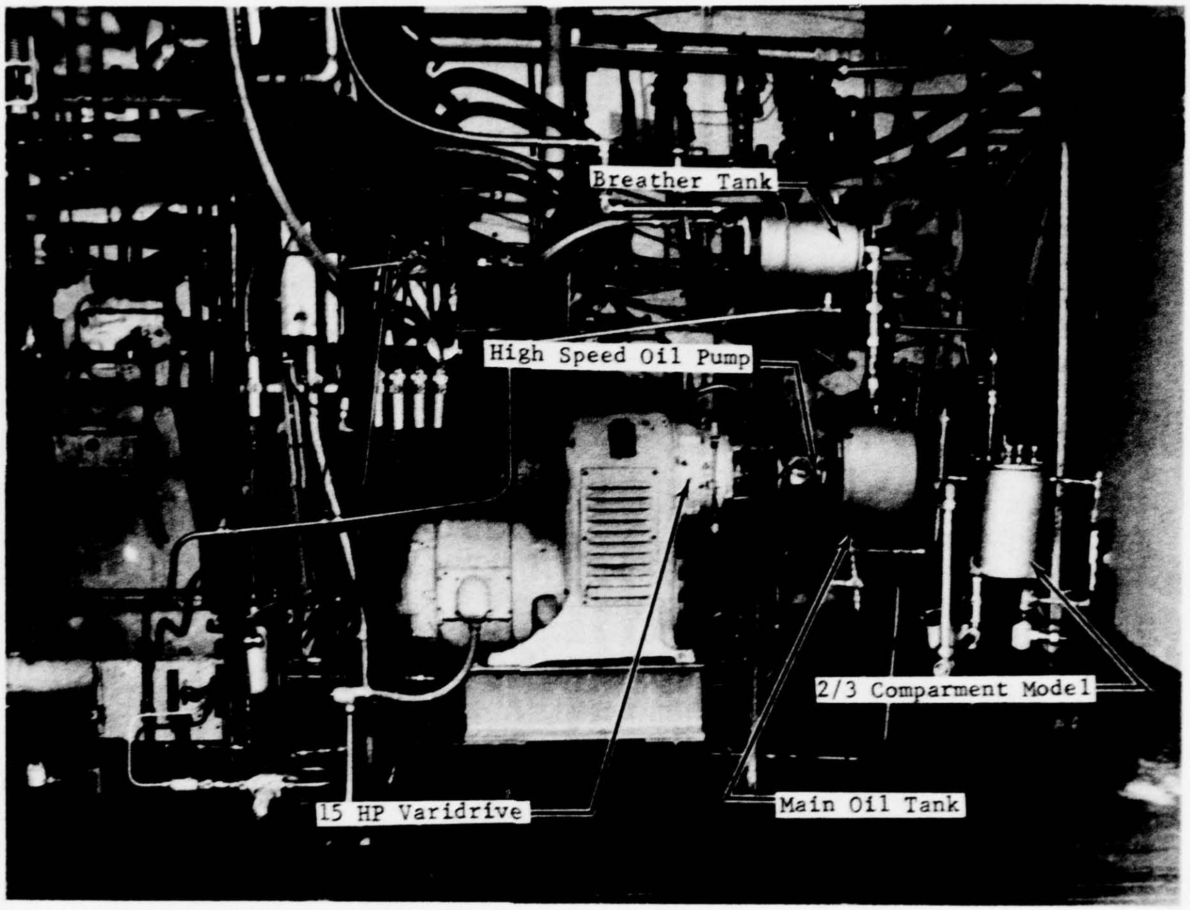
Figure 28 shows the oil supply pump lift capabilities at 10,000 rpm. Delivered oil flowrate is shown unaffected when oil levels in the tank were as much as 24 inches below pump inlet.

The operational curve for the cold start bypass valve is shown in Figure 29. This valve, an F100-PW-100 Bill-of-Material component, had an observed bypass threshold point at the design pressure differential of 175 psid.

Figure 30 is a pump map of the oil scavenge pump at 7000, 8500, and 10,000 rpm operating speeds. The Equipment Test Plan guarantee flowrate (shown superimposed in the figure) was surpassed at 10,000 rpm thus satisfying contractual flow requirements.

c. Oil Tank and Deaerator Performance

The compartmental oil tank, with a maximum capacity of 2.75 gallons, was injected with up to 200 lb/hr airflow with oil levels down to 1 gallon. Pressure oscillations of less than ± 3 psi (at supply pump discharge location) were observed when deaerating 200 lb/hr airflow with 1.5 gallons of oil in the tank. This is over three times conventional engine tank deaeration requirements. The deaeration capabilities of the compartmental oil tank at various oil fills and injected airflows are shown in Figure 31.



PC 49235

Figure 24. D-4 Stand Pump, Tank and Stand Plumbing

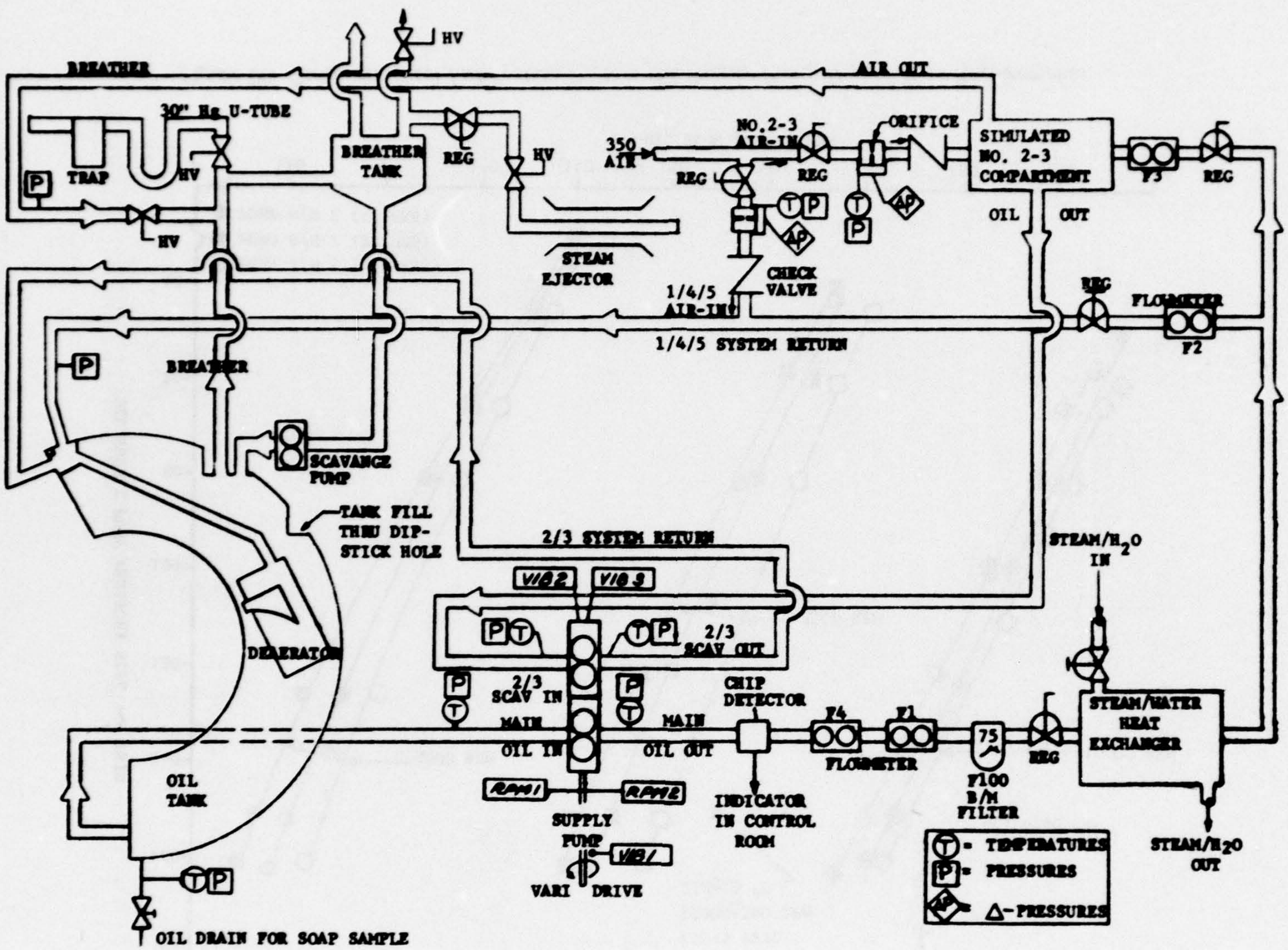


Figure 25. D-4 Stand Schematic

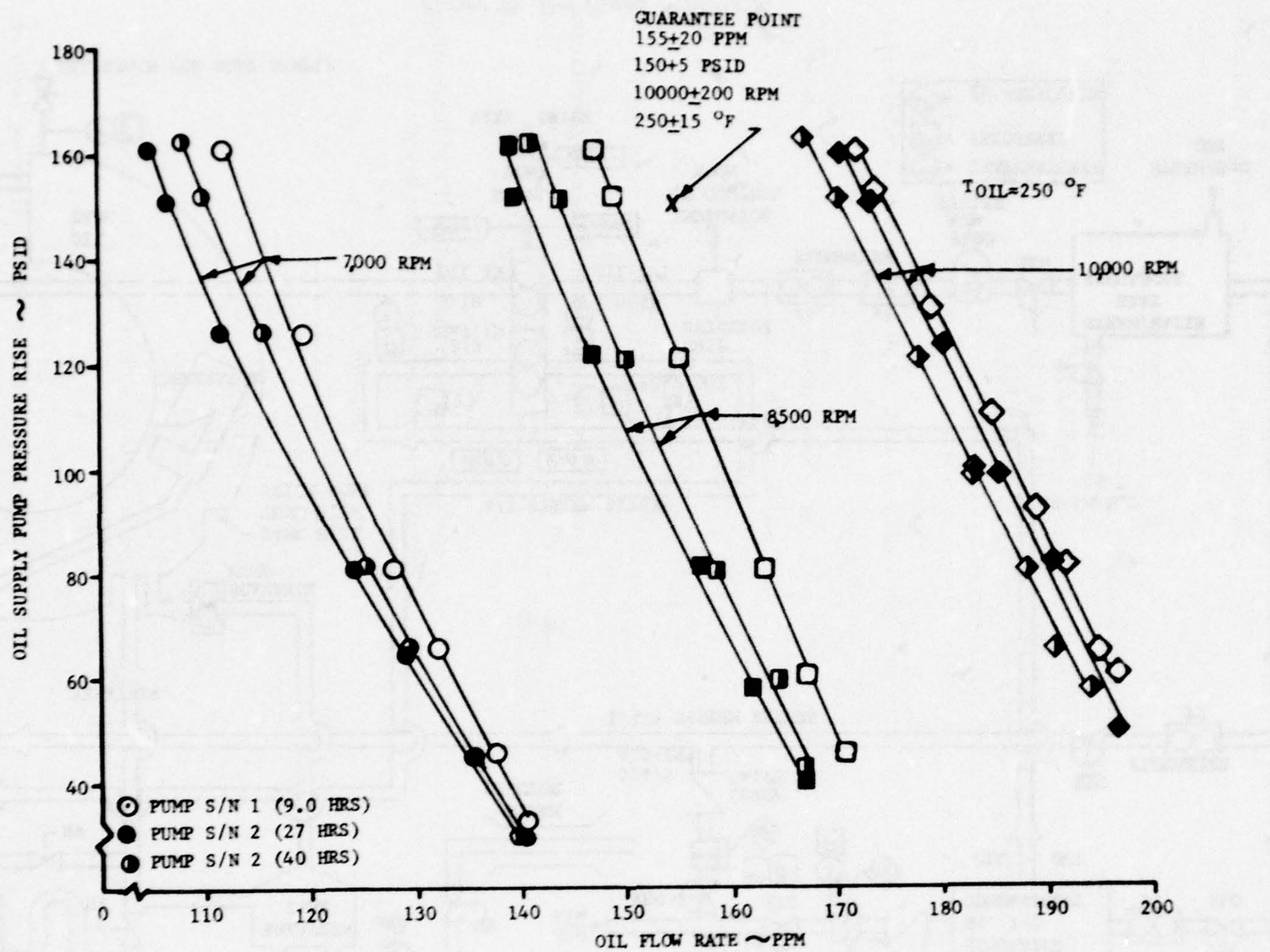


Figure 26. Compartmental Lubrication System Oil Supply Pump Design Flow Requirements

/02

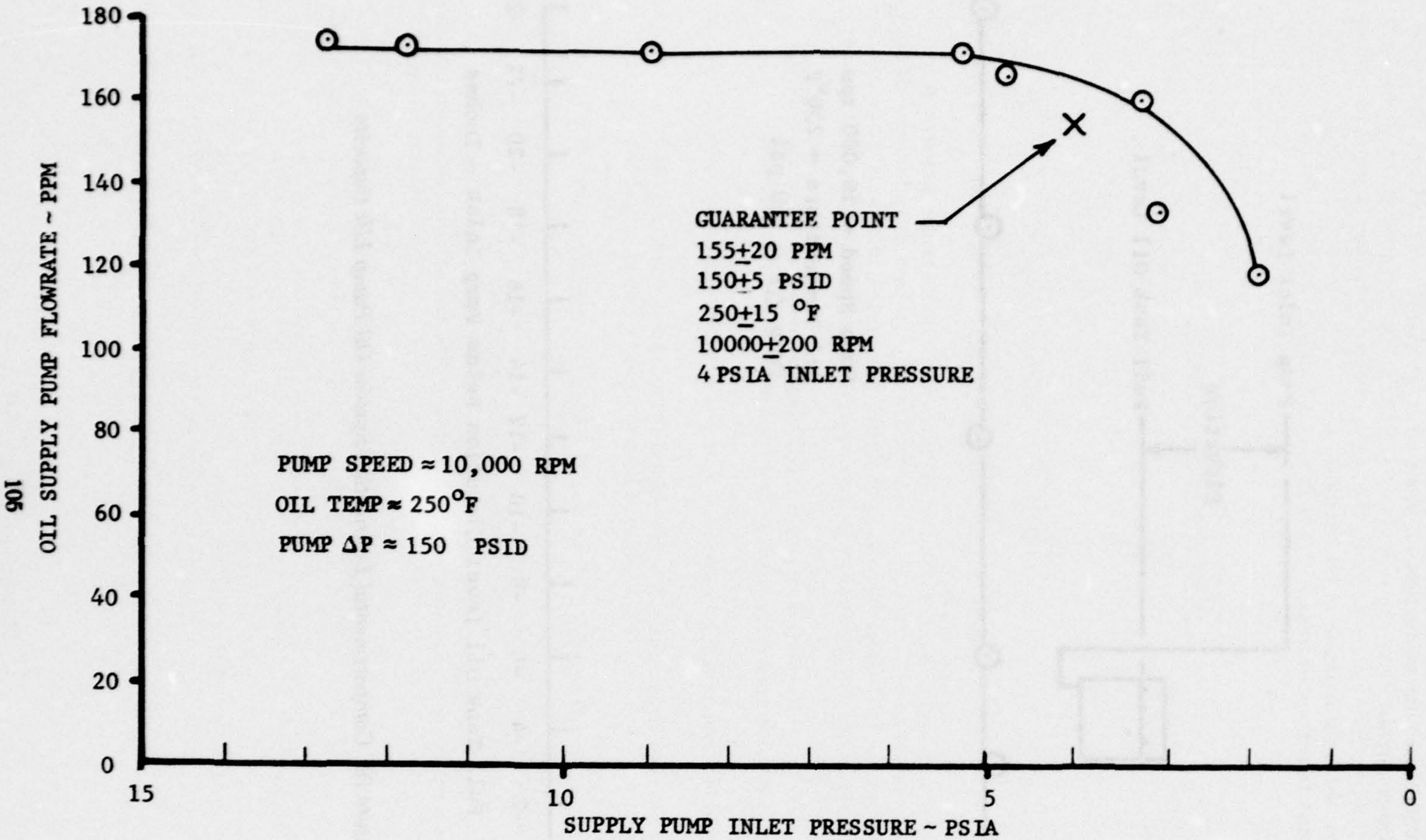


Figure 27. Compartmental Lubrication System Oil Supply Pump Flow Capacity at Low Inlet Pressures

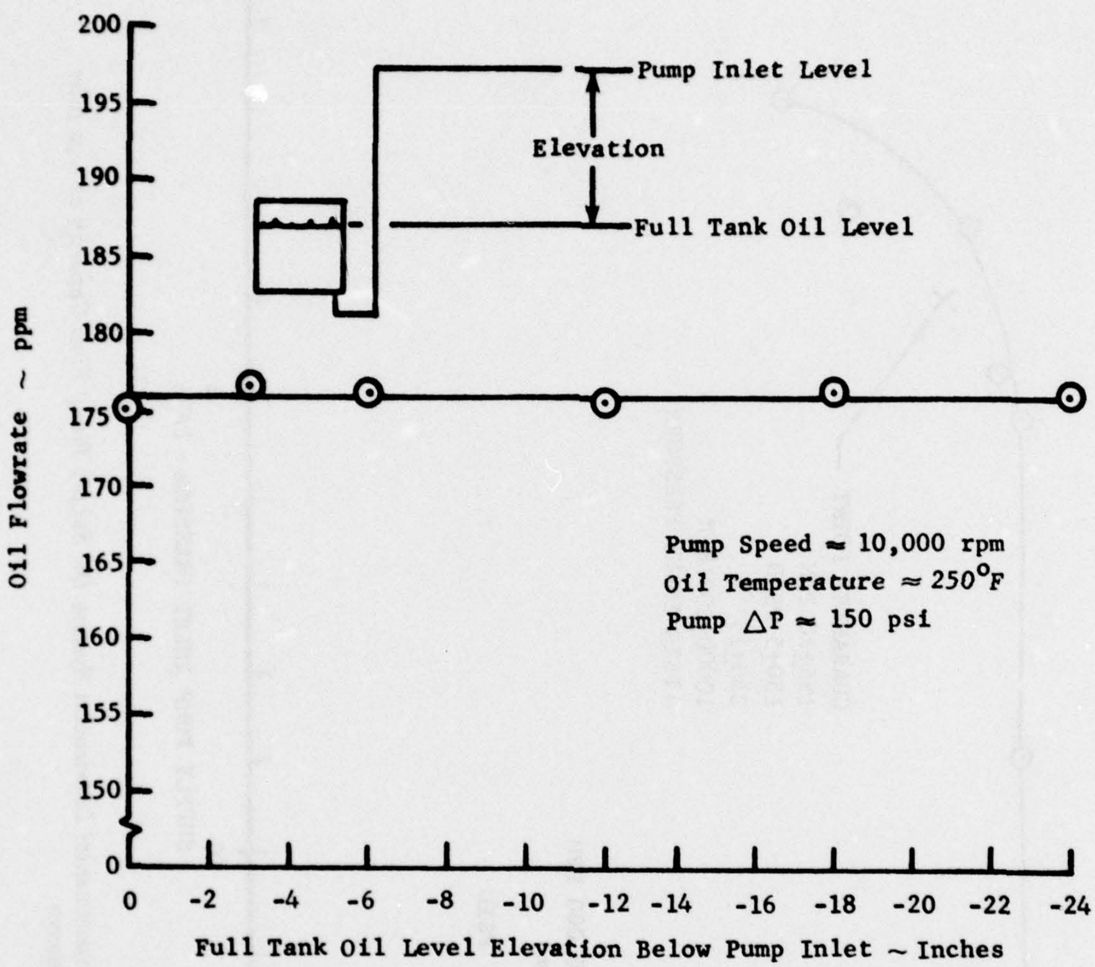


Figure 28. Compartmental Lubrication System Oil Pump Lift Capacity

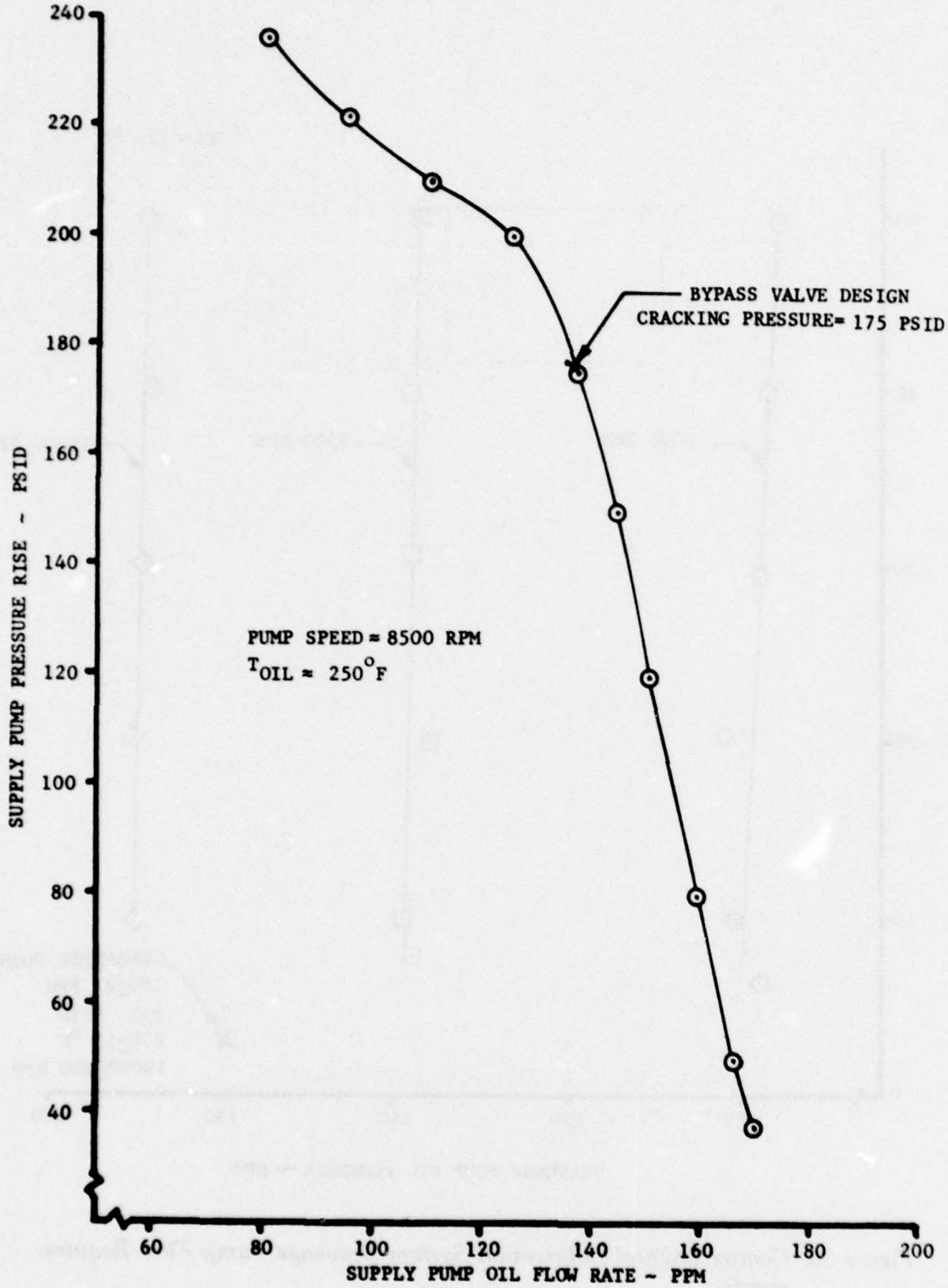


Figure 29. Compartmental Lubrication System Supply Pump Cold Start Bypass Valve Operation

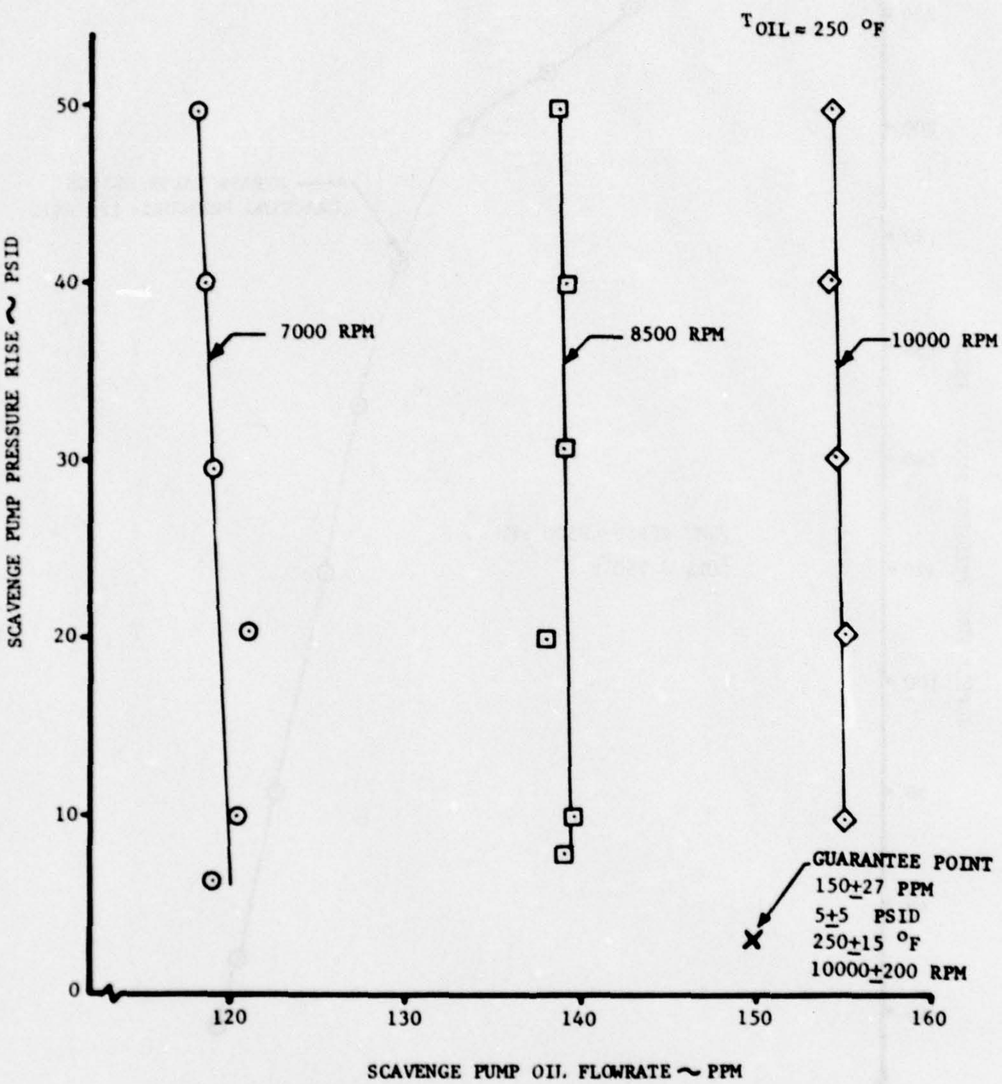


Figure 30. Compartmental Lubrication System Scavenge Pump Flow Requirements

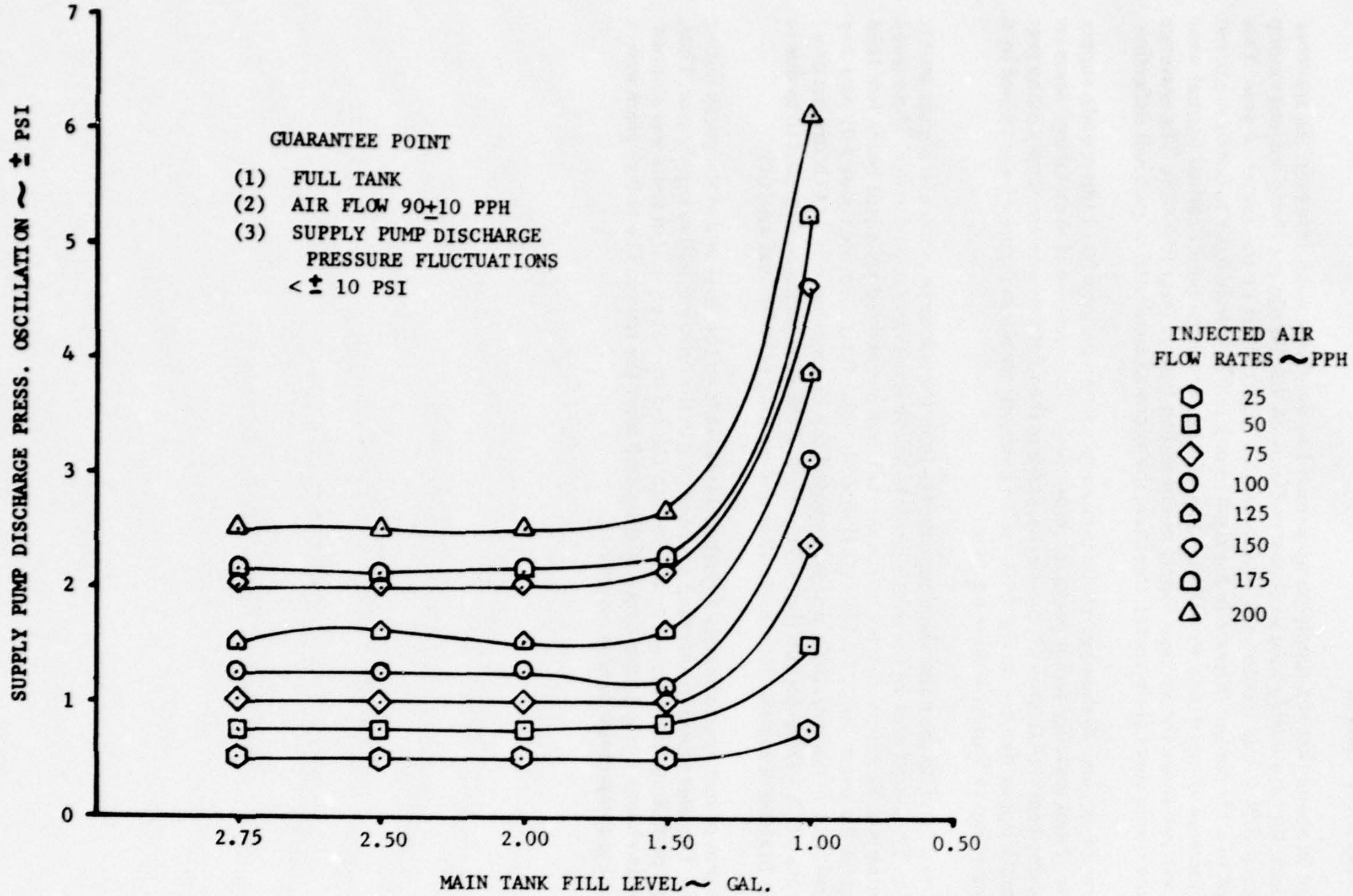


Figure 31. Compartmental Lubrication System Oil Tank Deaeration Capabilities Up to 200 lb/hr Airflow

d. Post-Test Observations

Figure 32 shows that the supply pump journal had no visible wear. However, the scavenge pump journal did have visible wear as shown in Figure 33. Radial run out of one scavenge pump gear in pump S/N 2 was 0.0015 inch over blueprint max (0.002) at the center of gear. This contributed to a 131 percent increase in backlash in pump S/N 2 (from 0.002 to 0.005) compared to S/N 1 increase (from 0.0045 to 0.0058). The average scavenge pump carbon journal wear (0.00034) was two times the average supply pump carbon journal wear (0.00018). The scavenge pump journals were very lightly loaded, therefore, the wear was attributed to gearshaft deflection.

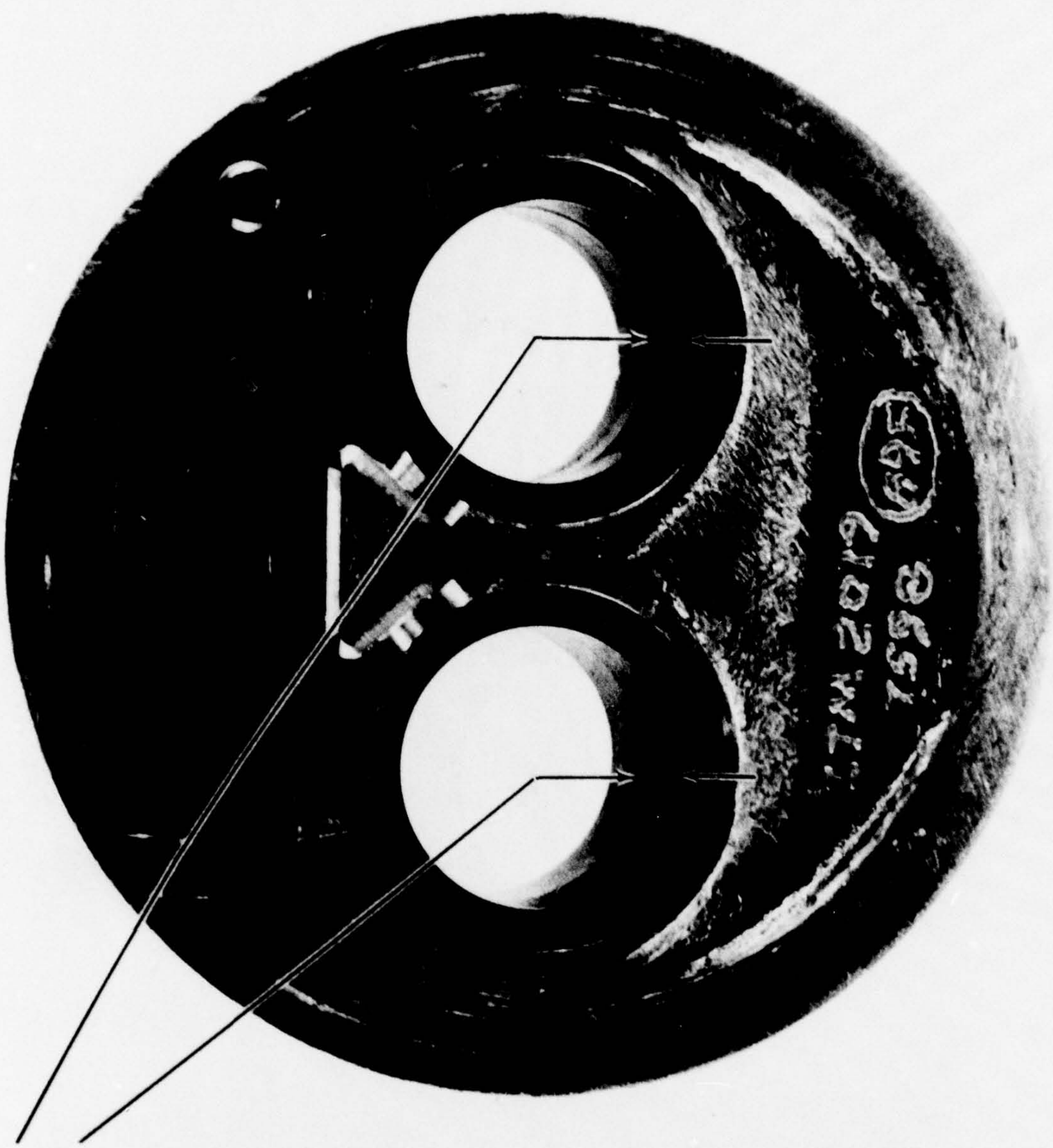
Figures 34, 35, and 36 show gearshafts from supply pump package No. 1 (drive end), supply package No. 2 and scavenge pump package, respectively. The presence of worn (shiny) areas on scavenge pump gear face (Figure 37) can be compared to the relatively unworn supply pump gear face Figure 38. Spline damage on the drive end of gearshaft shown in Figure 34 was caused by a loose fitting pump to Varidrive drive adaptor.

Figures 37 and 38 show the aluminum sleeves from the scavenge pump and supply pump, respectively. The pitted area on the inlet side of the sleeve is not cavitation damage. These small holes were caused by silicon spheres in the oil. Oil analysis showed spherical beads less than 60 microns in diameter. An F100-PW-100 Bill-of-Material filter was used and will pass any material less than 70 microns. Both pumps displayed this damage but S/N 2 (40 hours run time) was worse than S/N 1 (20 hours run time). The origin of the glass beads is suspected to be due to incomplete flushing of interior tank parts after grit blasting prior to final assembly.

Both pumps had approximately 2 qts/hr oil leakage from the drive end of the pump during initial tests. This was solved by inserting a rubber plug in the drive end hollow supply gear. Thus, the leakage path from the scavenge pump through the hollow supply pump gears was stopped. There was no change in the performance of the pump after the repair. The hollow gears were a manufacturing compromise to ease machining.

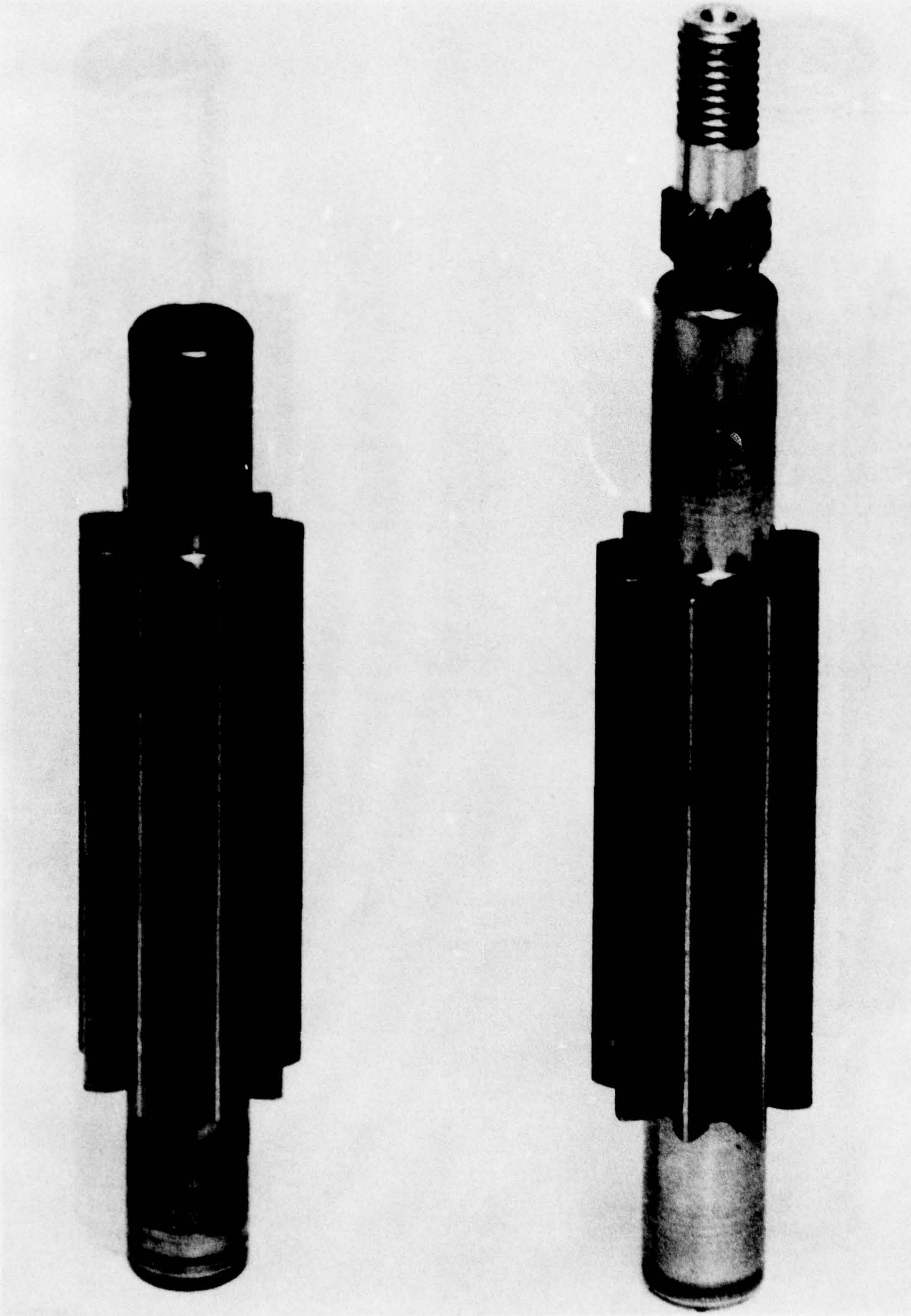
FE 158014

Figure 32. Supply Pump Bearing Assembly



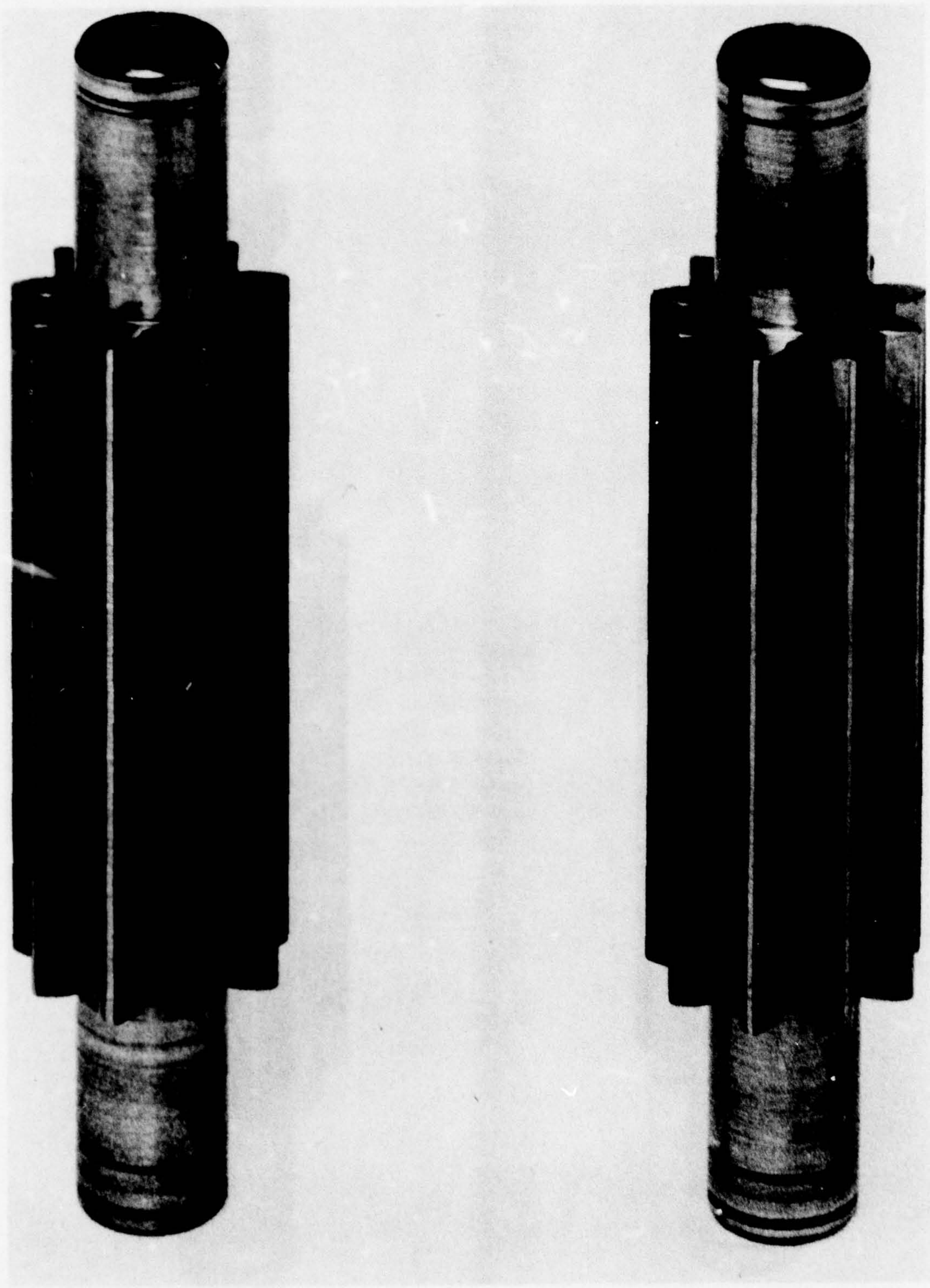
FE 157985

Figure 33. Scavenge Pump Bearing Assembly



FAE 157984

Figure 34. Supply Pump Gearshafts (Drive End), 40 Hours Run Time



FAE 157983

Figure 35. Supply Pump Gearshafts, 40 Hours Run Time

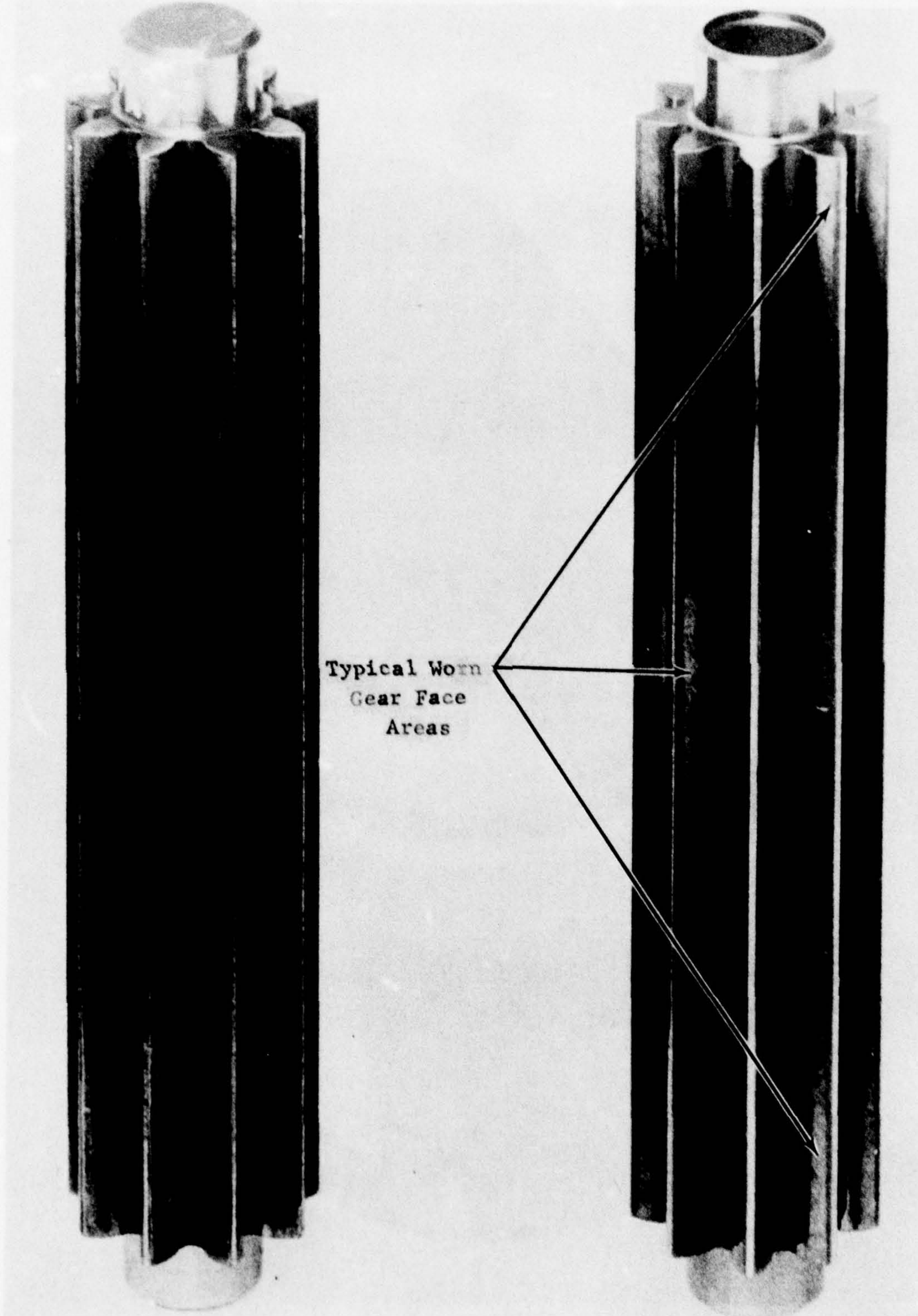
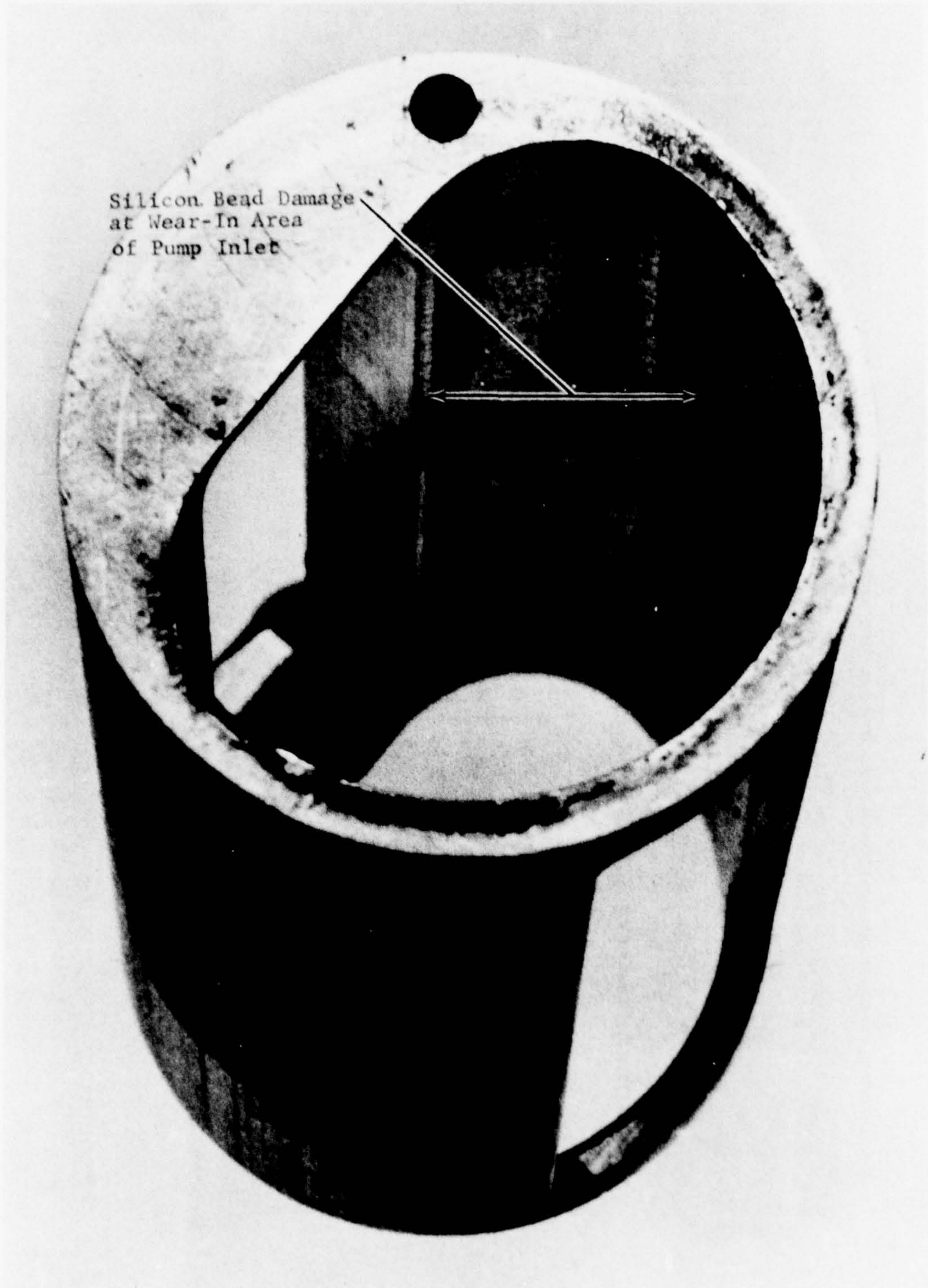
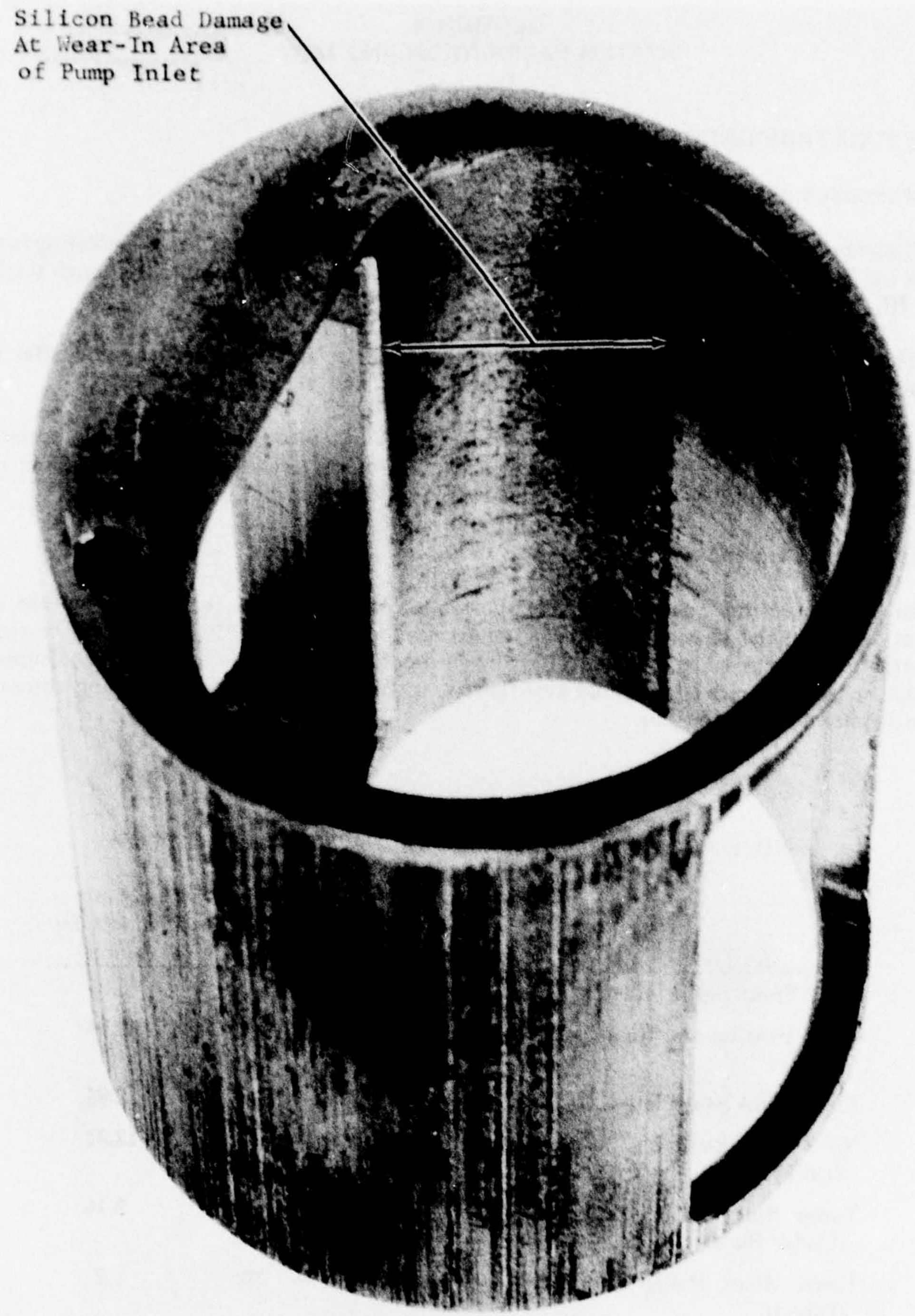


Figure 36. Scavenge Pump Gearshafts, 40 Hours Run Time



FE 158013

Figure 37. Scavenge Pump Sleeve Silicon Particle Damage



FAE 157981

Figure 38. Supply Pump Sleeve Silicon Particle Damage

SECTION V
SYSTEM FABRICATION AND TEST

1. SYSTEM FABRICATION AND ASSEMBLY

a. Description of Test Articles

High-speed oil supply and scavenge pumps S/N 1 (Figure 39) were selected for testing in the system rig. This pump had accumulated 20 hours run time during the component bench tests of Phase III, Task 2.

The compartmental tank (Figure 19) was flushed out and visually inspected after the critical component tests prior to its installation in the system rig.

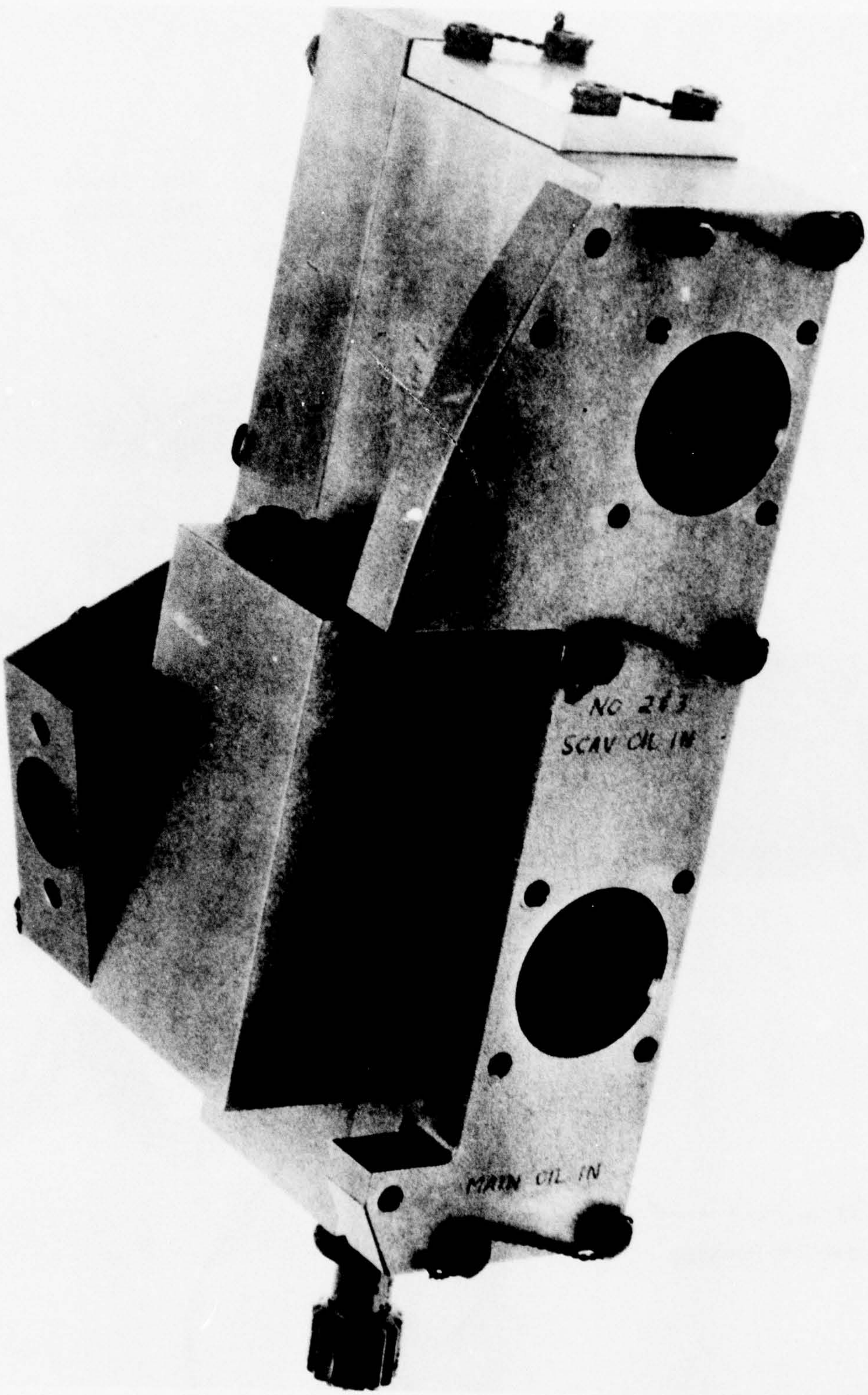
The high-speed gear drive described in Section IV was used to drive the high-speed oil supply and scavenge pumps. The high-speed gear train (Figure 40) was instrumented and assembled to the F100-PW-100 Bill-of-Material No. 2/3 crossover housing.

b. Assembly Sequence

In preparation for final assembly of the system rig there were several flow checks and reworks accomplished to ensure proper system operation. The F100-PW-100 No. 2/3 crossover support, No. 2 bearing oil supply, No. 3 bearing oil supply, No. 2 and No. 3 seal plate oil supplies and high-speed gear train oil manifold were flowed separately. All individual oil supply rates met design requirements (Table 25).

TABLE 25
NO. 2/3 COMPARTMENTAL LUBRICATION RIG OIL FLOW

Jet Location	*No. of Jets*	*Required Flow Per Jet (lb/min)*	*Actual Flow Per Jet (lb/min)*
No. 2 Front Seal Plate	1	4.5 — 6.0	5.2
No. 2 Bearing and Rear Seal Plate	1	16.0 — 19.0	20.54
No. 3 Front Seal Plate	3	2.0 — 3.0	2.83
No. 3 Bearing and No. 3 Rear Seal Plate	3	11.0 — 13.0	12.31
Tower Shaft Roller Bearing (Under Race)	1	2.5 — 3.5	3.16
Tower Shaft Roller Bearing (Direct)	1	0.5 — 1.5	1.2
Lower Tower Shaft Bearing and Idler Bearings (2)	3	1.5 — 4.5	2.72



FE 154782

Figure 39. High-Speed Supply and Scavenge Pump Assembly

Figure 40. High-Speed Gear Train

Operations sheets, guiding the inspection and assembly of the system rig, were generated to ensure proper assembly sequence and dimensional inspection during build up.

The F100-PW-100 No. 2 front, No. 2/3 and No. 3 rear carbon seal assemblies were lapped to a flatness of 0.000020 inch. The corresponding seal plates were inspected to assure 0.000020-inch flatness.

The rig was assembled using F100-PW-100 spiral wound crush gaskets in the static seal areas.

Prior to and during assembly, sufficient inspection and stack-up data were taken to assure seating of seal plates and proper compression of carbon seal assemblies.

The high-speed gear train was assembled; gear tooth alignment was checked, and backlash measurements were taken. Bull gear and pinion gear tooth contact pattern were checked prior to final assembly.

Figure 41 shows the interior of the system rig with the high-speed oil supply and scavenge pumps, compartmental tank, high-speed gear train, and all associated plumbing and instrumentation installed.

c. Rig Support Work

The high and low rotor of the system rig were driven commonly through a coaxial gearbox. The gearbox was overhauled and reworked to ensure proper operation.

The necessary tooling for assembly and disassembly of the rig was fabricated. Special tooling required for assembly and disassembly of the compartmental lubrication system components such as the high-speed gear drive was also fabricated.

Inspection of thrust piston knife-edge seals and lands revealed abnormally large radial clearance. Flowrate calculations based on these clearances revealed air flow requirements that exceeded facility capabilities. The knife-edges and lands were reworked to reduce the air flows required to obtain proper loads on the main shaft bearings.

d. Instrumentation Installation

All thermocouple, pressure and vibration instrumentation associated with the rig directly was patterned after requirements specified in the Equipment Test Plan. All internal rig thermocouples were shielded chromel alumel type installed through airtight fittings.

All internal rig pressure probes were inserted into their respective compartment to a depth that would give representative data for that parameter.

Dual bearing outer race thermocouples were installed in the supports for the No. 2, No. 3, upper towershaft, lower towershaft, upper idler, and lower idler bearings. These were flush mounted and in direct contact with the outer race outside diameter.

Numerous rig external pressure sensors and thermocouples were used to adequately monitor the operation of the rig, coaxial gearbox, and stand drive.

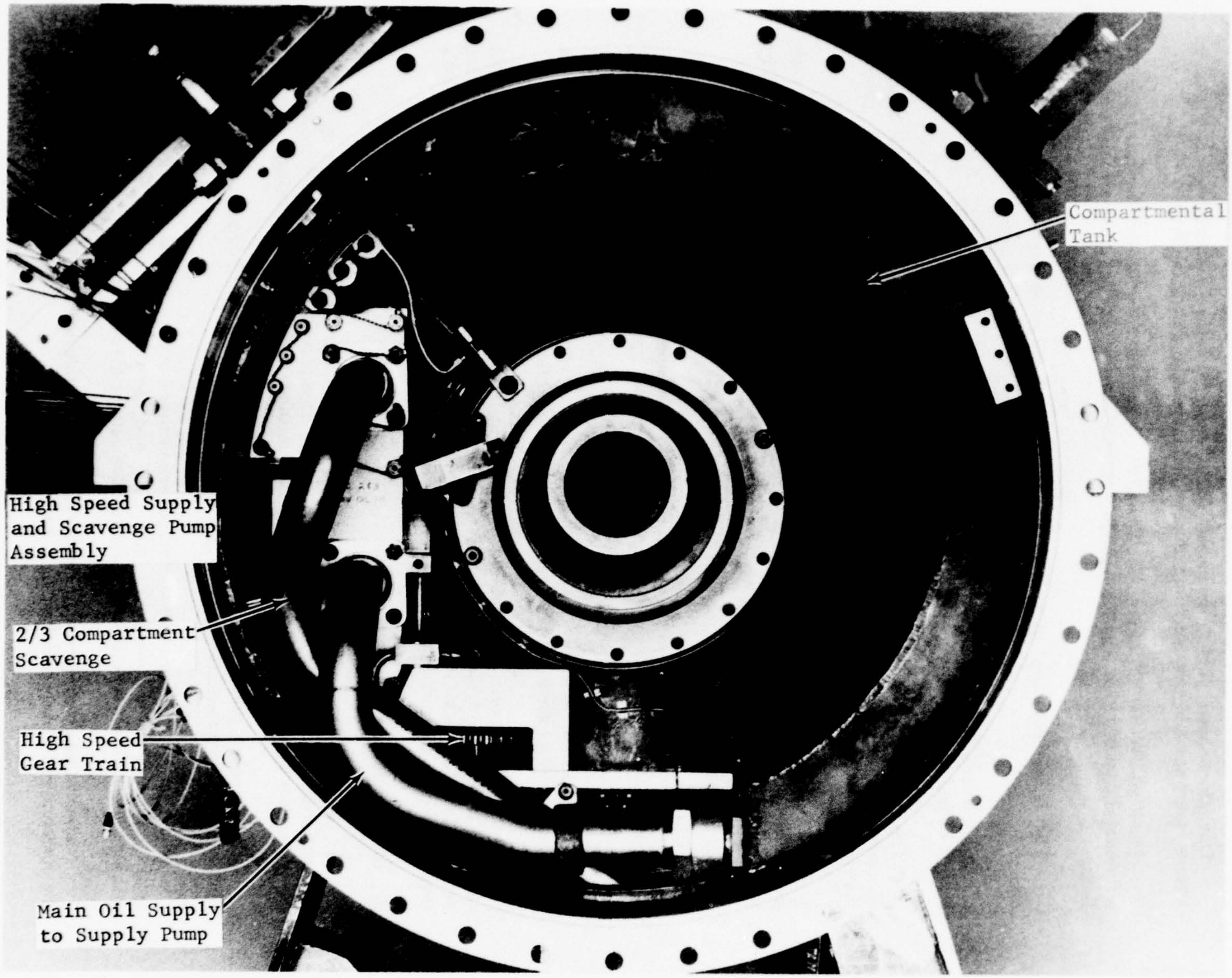


Figure 41. Compartmental Lubrication System Rig Interior

The instrumentation schedule is shown in Table 26. Vibration accelerometers were used to monitor rig internal and external vibrations. Internal sensors consisted of two radial accelerometers on the No. 3 bearing support and one radial accelerometer on the No. 2 bearing support. External sensors consisted of horizontal and vertical accelerometers on both front and rear of the main rig housing. Horizontal and vertical radial accelerometers were also installed on the coaxial gearbox.

2. SYSTEM TEST

The system rig was installed on D-4 stand, Turbojet Component Test Area D. The drive system for the rig was a 250 hp Ford V8 engine. Power was supplied to the rig through a torque converter, 5-speed manual transmission and reversing gearbox. A drive shaft connected the stand drive system to the rig coaxial gearbox.

Slight modifications were required to the rig mount stand to adapt it to D-4 stand. Stand, rig, and all plumbing are shown in Figure 42.

The rig and all air inlet lines were insulated with fiber insulation, aluminum foil, and fiberglass tape to reduce heat loss. Control room instrumentation consisted of pressure gages, digital thermocouple readouts, digital speed and oil flowrate readouts, and vibration level meters.

Disaster monitoring was accomplished by using an o'graph recorder. Seven channels were recorded which included two bearing temperatures, two rig vibrations, high-rotor speed, No. 2/3 compartment breather pressure, and No. 2/3 compartment oil supply pressure. The purpose of disaster monitoring selected parameters was to have a record of rig operating characteristics in the event of rig malfunction since hand-recorded data would not be fast enough.

Figure 43 shows the stand schematic for the system rig installed on D-4 stand.

Oil flowrates were measured with calibrated turbine flowmeters. Standard sharp-edged orifices were used for measuring air flows to the various chambers and cavities in the rig.

Rig bearing thrust loads were controlled by setting thrust piston pressure differential. Thrust balance calculations were completed for both front and rear thrust pistons for each mission point.

During the rig checkout period prior to beginning the endurance run, high breather air flow was noted. By flowing each compartment separately, the leak was found to be in the area of No. 3 rear seal. This prevented setting the rear chamber pressures required for the climb and combat mission points. Repair would have required a complete dismount and teardown. Since the leakage did not affect the test article, i.e., high-speed oil supply and scavenge pumps, compartmental tank, and high-speed gear train, it was decided to continue the endurance test with reduced rear chamber pressures. Chamber temperatures for each mission point were met with no problems.

It was apparent, while setting the mission points during the checkout runs, that the amount of time required to set the oil flow, air flows, oil and air temperatures, compartment pressures, breather pressure, and rig speed was not conducive to a cyclic test. Approximately three hours were required to set a point so that the operation of the rig during transients was really not being evaluated. The critical items for the system test (i.e., operation of the high-speed pump and drive train, oil churning in a compact bearing compartment, and deaeration capabilities) could all be thoroughly evaluated at steady-state operating conditions. It was decided to combine the test times for each mission flight point and revise the test sequence as shown in Table 27. Note that the low-power points were run first. The facilities drive engine for the rig was found to be defective during the checkout runs and was replaced with a new drive engine. The low-power points were run first to help break in the engine.

TABLE 26
COMPARTMENTAL LUBRICATION SYSTEM INSTRUMENTATION SCHEDULE

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

INSTRUMENTATION SCHEDULE
EXPERIMENTAL TEST DEPARTMENT

Sheet 1 of 2
Original Date 11/28/77
Revised Date

Engine/Rig No. F34024
Stand D-4
Work Order No. 4163-03-012-XX

Type 2/3 Compt. Rig (RL10, JT11, Etc)
Build No. 10
Run Date 1/6/78

Test of Compartmental Lube System
Test Engineer Bill Gamble Ext 3239
Alt Test Engineer Dave Smith Ext 3250

No.	Item Description	Header	Expected Range	Units	Environment	TC Type	Gage	ADR	O-Graph	Meter	Strip Charts	Remarks
							Recording and Readout Required					
1	TEMPERATURES											
2												
3	Oil Tank Temp	T1	AMB-500	°F	Oil	C/A	Doric					Installed at Test in Plumbing
4	Oil Supply Pump Discharge Temp	T2										" " " " "
5	No. 2/3 Compartment Supply Temp	T3										" " " " "
6	No. 1/4/5 Oil Supply Temp	T4										" " " " "
7	No. 2/3 Compartment Air Temp	T51			Oil/Air							Female Connector Rig Stand Off
8	" " " " "	T52			Oil/Air							" " " " "
9	No. 1/4/5 Air Orifice	T6			"							Orifice S/N 180.1
10	Forward Chamber (Cavity A) Temp	T71	AMB-750		Air							Female Connector Rig Stand Off
11	" " " "	T72										" " " " "
12	Rear Chamber (Cavity B) Temp	T81										" " " " "
13	" " " "	T82										" " " " "
14	Bore (Cavity C) Temp	T9										Installed at Test in Plumbing
15	Forward Dome Temp	T10										Female Connector Rig Stand Off
16	" " "	T12										" " " " "
17	Rear Dome Temp	T11										" " " " "
18	" " "	T13										" " " " "
19	Breather Air Orifice Temp	T15	AMB-500									Orifice S/N 1205.1
20	No. 2/3 Scav Pump Disc Temp	T16			Oil							Female Connector Rig Stand Off
21	Bore Air Out Temp	T14			Air							Installed at Test in Plumbing
22	Forward Dome Air Orifice Temp	T17	AMB-750									" " " " " S/N 954.1
23	Rear Dome Air Orifice Temp	T18	"									" " " " " S/N 786.0
24	Gearbox Oil In Temp	T19	AMB-250		Oil							" " " " "
25	Gearbox Oil Out Temp	T20	"		"							" " " " "
26												
27												
28												
29	BEARING TEMPERATURES											
30												
31	No. 2 Brg	B11	AMB-500	°F	Oil	C/A	Doric					Female Connector Rig Stand Off
32	No. 2 Brg	B12							X			
33	No. 3 Brg	B21										
34	No. 3 Brg	B22							X			
35	Pump Drive Upper Idler Brg	B31										
36	" " " " "	B32										
37	Pump Drive Lower Idler Brg	B41										
38	" " " " "	B42										
39	Lower Towershaft Ball Brg	B51										
40	" " " "	B52										
41	Upper Towershaft Roller Brg	B61										
42	" " " "	B62										
43												
44												
45												
46												

TABLE 26
COMPARTMENTAL LUBRICATION SYSTEM INSTRUMENTATION SCHEDULE (Continued)

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

INSTRUMENTATION SCHEDULE
EXPERIMENTAL TEST DEPARTMENT

Sheet 2 of 2
Original Date 11/28/77
Revised Date

Engine/Rig No. F34026
Stand D-4
Work Order No. 4163-03-012-XX

Type 2/3 Compt. Rig (RL10, JT11, Etc)
Build No. 10
Run Date 1/6/78

Test of Compartmental Lube System
Test Engineer Bill Gamble Ext 3239
Alt Test Engineer Dave Smith Ext 3250

No.	Item Description	Header	Expected Range	Units	Environment	TC Type	Recording and Readout Required: Gage	ADH	O-Graph	Meter	Strip Charts	Remarks
1	PRESSURES											
2												
3	Oil Tank Pressure	P1	6-25	PSIA	Oil		Heise					Installed at Test in Plumbing
4	Oil Supply Pump Discharge Press	P2	6-200									" " " " "
5	No. 2/3 Compartment Supply Press	P3	AMB-100						X			" " " " "
6	No. 1/4/5 Oil Flow Press	P4	AMB-100									" " " " "
7	No. 2/3 Compartment Breather Press	P5	6-25		Air/Oil				X			" " " " "
8	No. 1/4/5 Air Orifice Press	P6	AMB-200		Air							Orifice S/N 180.1
9	Forward Chamber (Cavity A) Press	P7	AMB-100		Air							Rig Stand Off
10	Rear Chamber (Cavity B) Press	P8	AMB-100		Air							Installed at Test
11	Bore Chamber (Cavity C) Press	P9	AMB-100		Air							Installed at Test in Plumbing
12	Forward Dome Press	P10	AMB-200		Air							Rig Stand Off
13	Rear Dome Press	P11	AMB-200		Air							Rig Stand Off
14	Bore Air Exit Press	P14	AMB-250		Air							Installed at Test in Plumbing
15	Breather Air Orifice Press	P15	6-25		Air/Oil							Orifice S/N 1205.1
16	No. 2/3 Scav Pump Disc Press	P16	AMB-75		Oil							Rig Stand Off
17	No. 1/4/5 Air Orifice Delta Press	DP6	0-80	Inches Hg	Air		Hg Mano					Orifice S/N 180.1
18	Breather Air Orifice Delta Press	DP15	0-80	" "	Air		" "					Orifice S/N 1205.1
19	Forward Dome Air Orifice Delta Press	DP17										Orifice S/N 954.1
20	Rear Dome Air Orifice Delta Press	DP18										Orifice S/N 946.0
21	Forward Dome Air Orifice Press	P17	AMB-250	PSIA			Heise					Orifice S/N 954.1
22	Rear Dome Air Orifice Press	P18	" "	PSIA			"					Orifice S/N 916.0
23	Gearbox Oil In Press	P19	" "	PSIG	Oil		"					
24	Gearbox Oil Out Press	P20	" "	PSIG	Oil		"					
25												
26												
27												
28												
29												
30	OIL FLOW RATES											
31	Oil Supply Pump	F1	0-250	PPM	Oil	Digital Counter						S/N 3/4 PWP 18
32	No. 1/4/5 Compartment	F2	0-150			"						S/N 3/4 C 15683
33	No. 2/3 "	F3	0-150			"						S/N 1.0 C 15699
34	Gearbox	F4	0-20			"						S/N
35	VIBRATIONS											
36	No. 3 Brg 350° Radial	VIB 1	0-10	MILS	Oil				X	X		
37	No. 3 Brg 260° Radial	VIB 2							X	X		
38	No. 2 Brg Vertical Radial	VIB 3								X		
39	Front Rig Case Vertical Radial	VIB 4			Air					X		
40	Front Rig Case Horz Radial	VIB 5								X		
41	Rear Rig Case Vertical Radial	VIB 6								X		
42	Rear Rig Case Horz Radial	VIB 7								X		
43	Coaxial Gearbox Radial VERTICAL	VIB 8								X		
44	" " " HORZ	VIB 9								X		
45	Oil Pump SPEED	RPM Pump	0-10,000	RPM	Oil		Digital Display					46 Tooth Gear Inside Rig
46	High Rotor SPEED	RPM -[illegible]	0-13,000	RPM	Air		"		X			60 Tooth Gear On Stand Drive

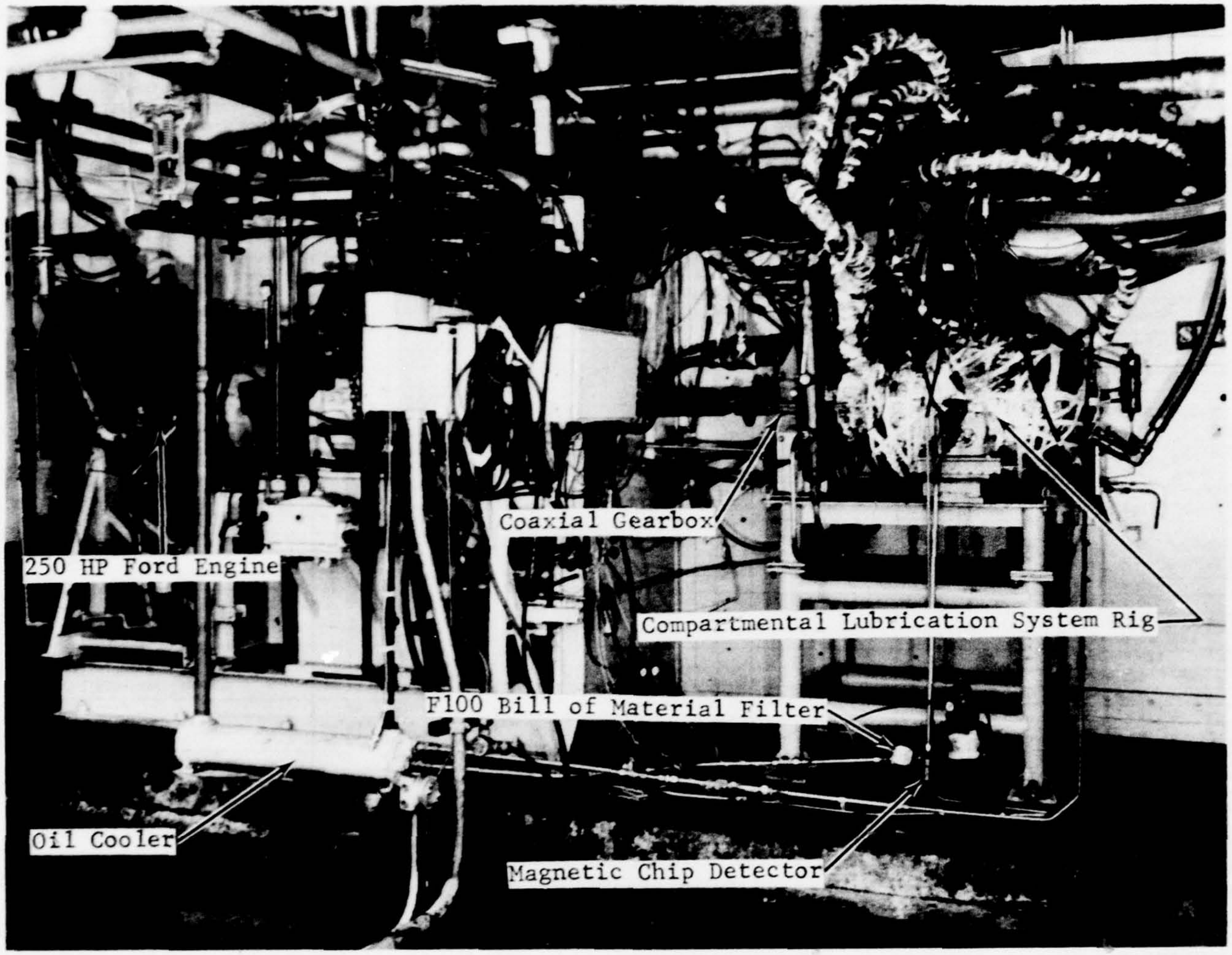


Figure 42. Compartmental Lubrication System Rig F34024-10 Installed on D-4 Stand

Plumbing Summary

Pipe No.	Quantity	Flow
A	1	Air Out
B	1	Air In
C	1	Air Out
D	1	Air/Oil Out
E	2	Air Out
F	2	Air In
G	2	Air In
H	8	Air Out
I	4	Air In

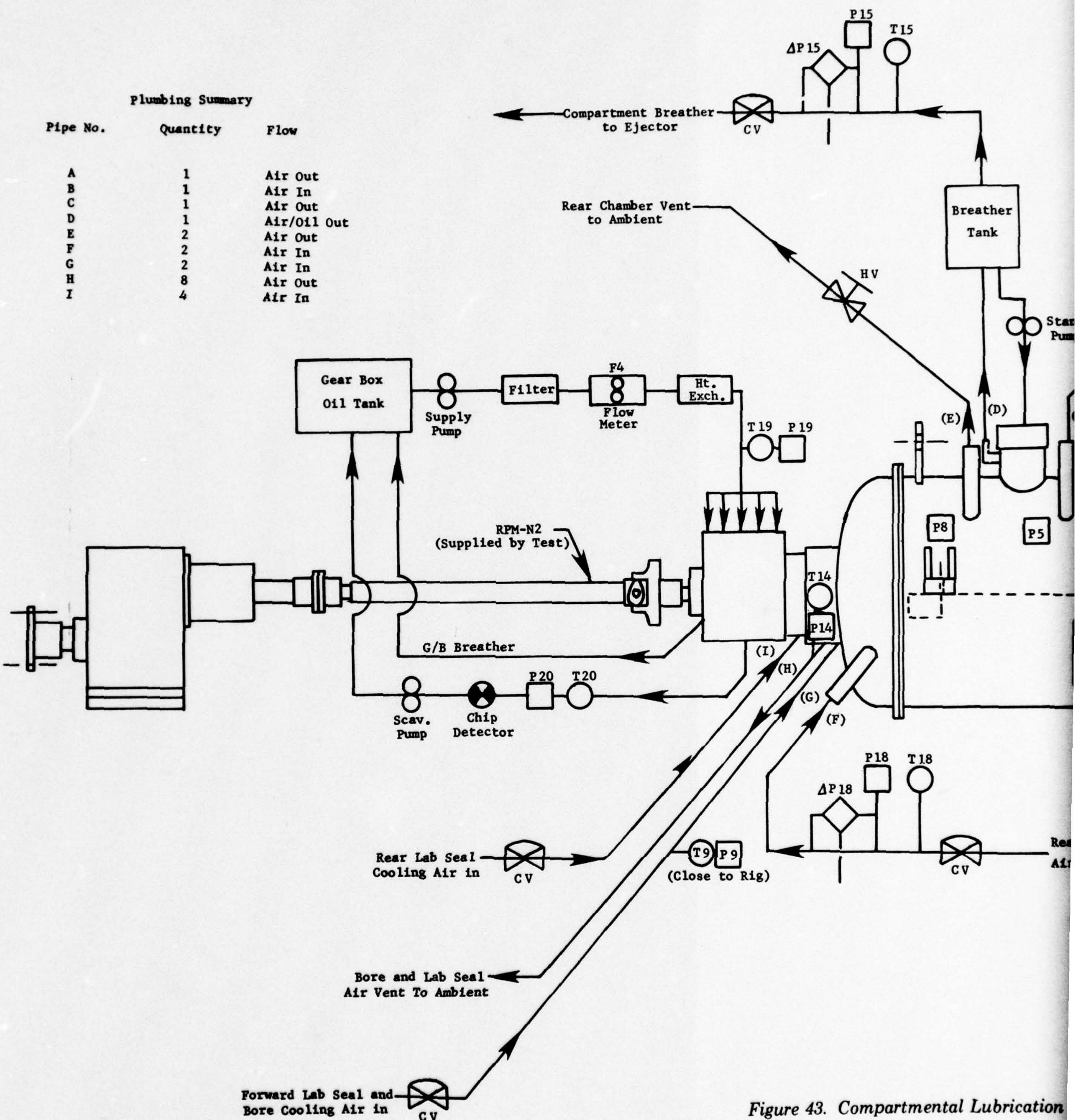
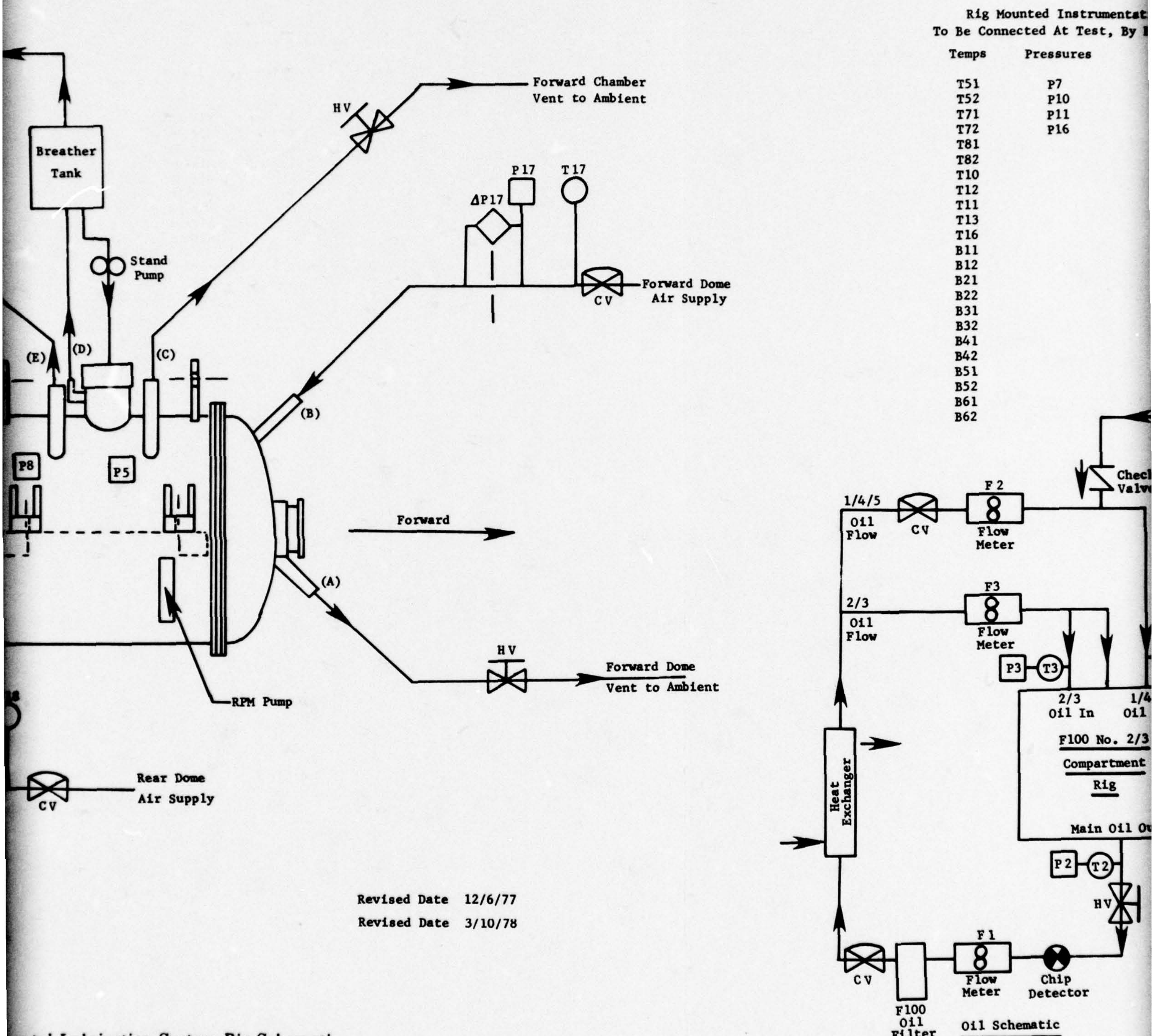


Figure 43. Compartmental Lubrication



ental Lubrication System Rig Schematic

2

Rig Mounted Instrumentation
To Be Connected At Test, By Headers

Temps	Pressures	Vibes
T51	P7	Vib-1
T52	P10	Vib-2
T71	P11	Vib-3
T72	P16	Vib-4
T81		Vib-5
T82		Vib-6
T10		Vib-7
T12		Vib-8
T11		
T13		
T16		
B11		
B12		
B21		
B22		
B31		
B32		
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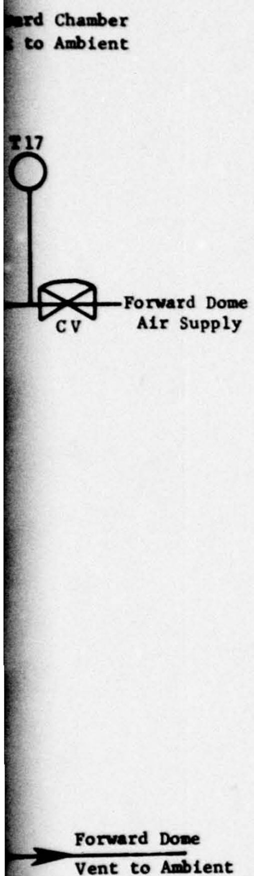
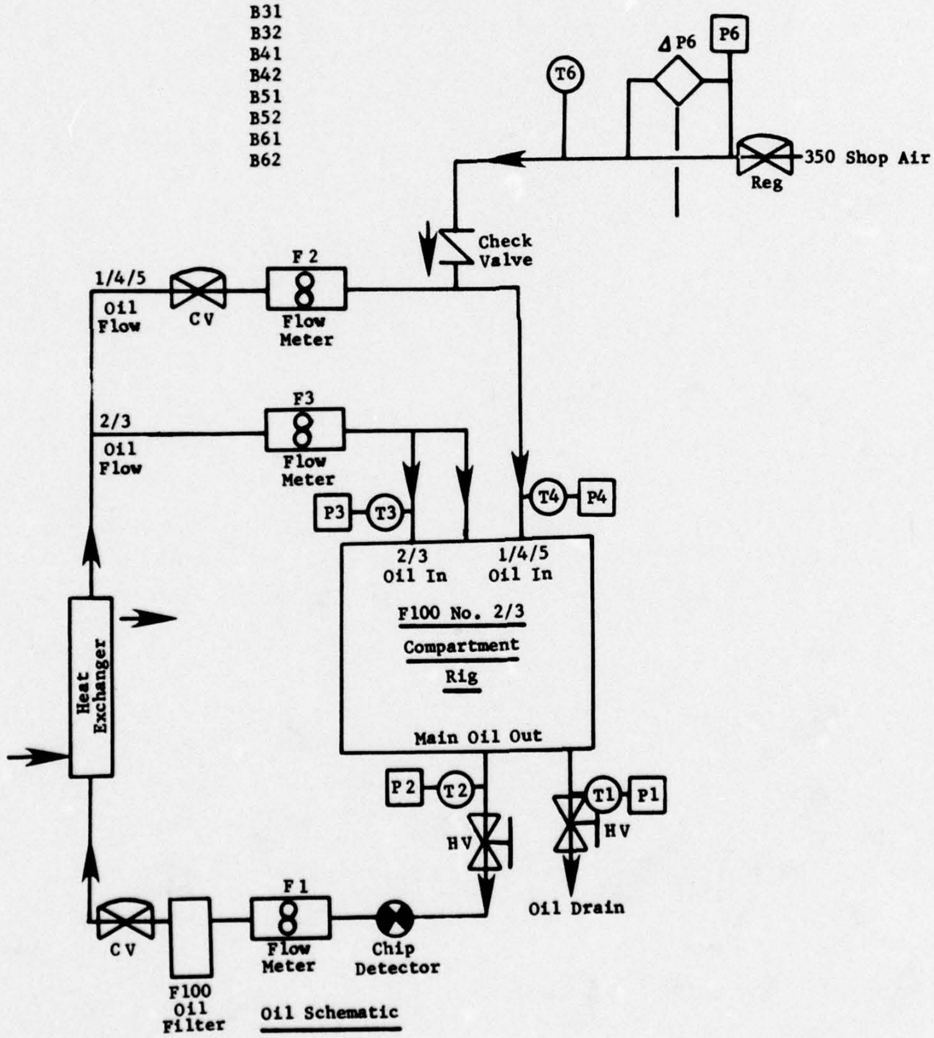
B51		
B52		
B61		
B62		





Oil Schematic

3

Spectrometer Oil Analysis and Processing (SOAP) samples were taken at 2.75, 6.75, 16.0, 20.0, 22.0, 28.0, 42.0, 46.0, and 50.23 hours of endurance time. Chip detectors were checked prior to every run and any collected material sent for analysis when required.

All data sheets from the endurance run are shown in Appendix O.

3. SYSTEM TEST RESULTS

a. System Rig Endurance Test Results

Total system rig run time at the end of 50.23-hours endurance time was 66.96 hours.

Figures 44 and 45 show the instrumentation locations. Figure 44 is a longitudinal cross section, and Figure 45 is a transverse cross section.

The bottom plot of each of the following graphs has rig high-rotor speed (RPM-N2) versus endurance time for reference. Each graph that has test set conditions plotted is labeled with the set point value. These values are listed in Table 27. All data were recorded by hand. In a few instances the set point drifted during data recording and is labeled on plots as transient.

Figure 46 shows high-speed oil pump speed (RPM-PUMP), No. 2/3 compartment oil supply pressure (P3), and main oil supply pump discharge pressure (P2), versus endurance time. It can be seen there was no deterioration of oil pressure with endurance time.

Figure 47 shows high-speed oil supply total oil flowrate (F1), oil tank temperature (T1), and No. 2/3 compartment oil flowrate (F3) versus endurance time. Again, neither parameter deteriorated with endurance time.

Figure 48 shows system rig compartmental heat generation and is based on No. 2/3 oil flowrate, oil supply temperature to the No. 2/3 compartment (T3), and No. 2/3 compartment oil scavenge temperature (T16). T3 and T16 are shown in Figures 48 and 49 with their difference (T16-T3) shown in Figure 50. Figure 48 also shows heat transferred in the heat exchanger as a check on the heat generation. This is based on average rig oil supply temperature (No. 2/3 and No. 1/4/5 compartment model), main oil supply discharge temperature (T2) and total rig oil flowrate (F1). Data scatter can be attributed to sensitive operation of water operated oil heat exchanger.

No. 2/3 compartment temperature rise (T16-T3) shown in Figure 50 was significantly lower than predicted. It is theorized that this is due to the absence of a towershaft in this rig. A significant reduction in heat generation may be realized in an engine with a top mounted gearbox due to the elimination in oil churning in the towershaft.

Figure 49 shows No. 2/3 compartment breather pressure (P5). Test set points are shown with the maximum allowable limits. Maximum allowable limits are 8 inches Hg (approx 4 psi) above the set point. Due to high breather air flow caused by the air leak in the rear of the No. 3 compartment, breather pressure was slightly higher than the set point at climb conditions. Figure 49 also shows No. 2/3 compartment scavenge pressure (P16). Scavenge pressure was measured at the high-speed scavenge pump discharge and did not fluctuate during any mission point.

TABLE 27
REVISED SYSTEM TEST POINTS

					Rotor Speed		Bearing Loads			Compartment Pressures and Temperatures				Nos. 1, 4, and 5	Simulated No. 2/3 Compartment	Estimated No. 2/3 Compartment
										"Cavity "A""		"Cavity "B" and "C""				
Flight Point	Condition	Time at Point, min	CUM Time, hr	Oil Supply Temperature, °F	High N_2, rpm	Low N_1, rpm	No. 2 Bearing, lb	No. 3 Bearing, lb	Breather Pressure, psia	Pressure, psia	Temperature, °F	Pressure, psia	Temperature, °F	Compt Seal Air Leakage Rate lb/hr	Oil Flowrate lb/min	Oil Temp Rise (Supply to Discharge) °F
1	Sea Level Idle	465	7.75	207	9140	6581	848	1503	14.7	15	136	18	195	46	56	30±5
6	Sea Level Idle	496	16.02	233	9140	6581	848	1503	14.7	15	136	18	195	46	56	30±5
3	Cruise Out	961	32.03	251	10912	7857	1721	3050	6.8	18	234	25	349	56.6	67	38±5
5	Cruise Back	837	45.98	251	10912	7857	1721	3050	6.8	18	234	25	349	56.6	67	38±5
4	Combat	155	48.57	196	12909	9295	4181	7410	8.3	29	408	55	568	160	79	82±10
2	Climb	93	50.12	192	13009	9367	5243	9291	12.2	16	429	69	588	200	80	90±10

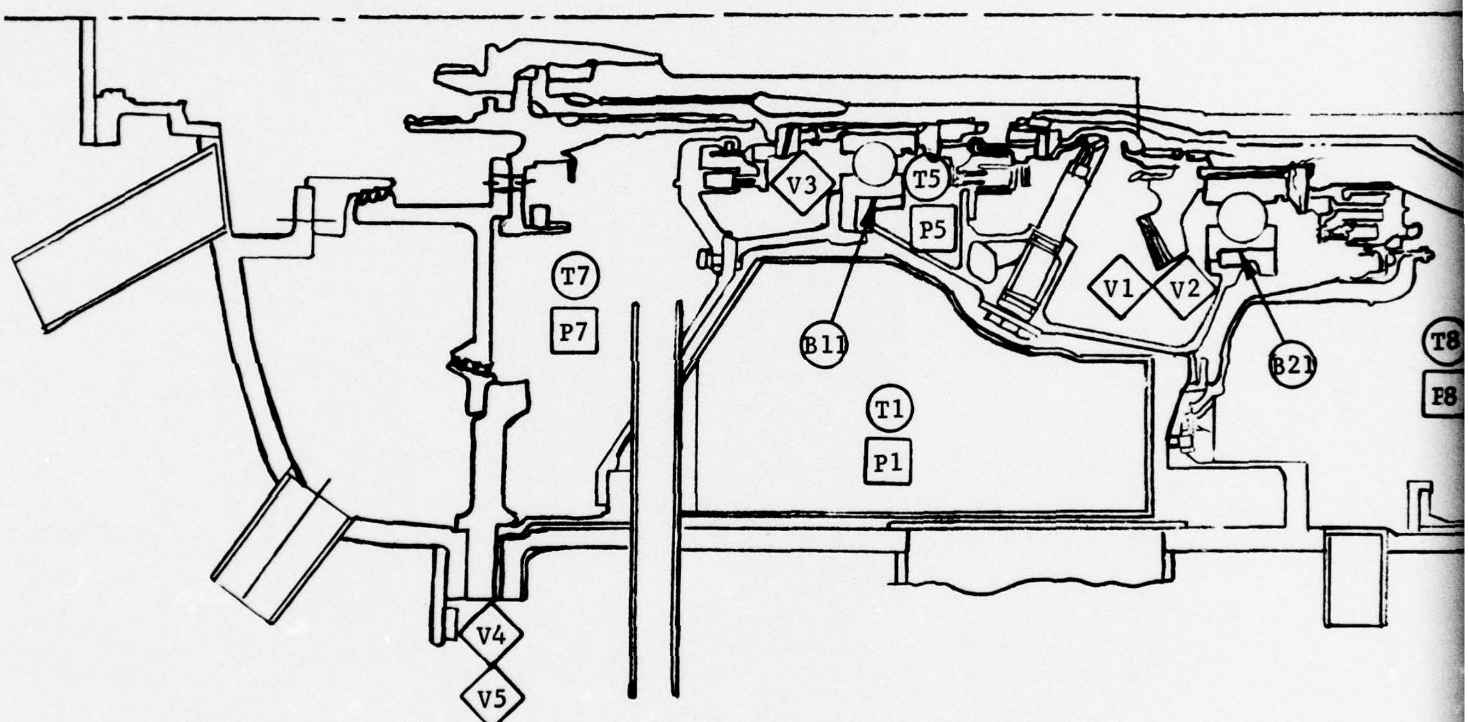
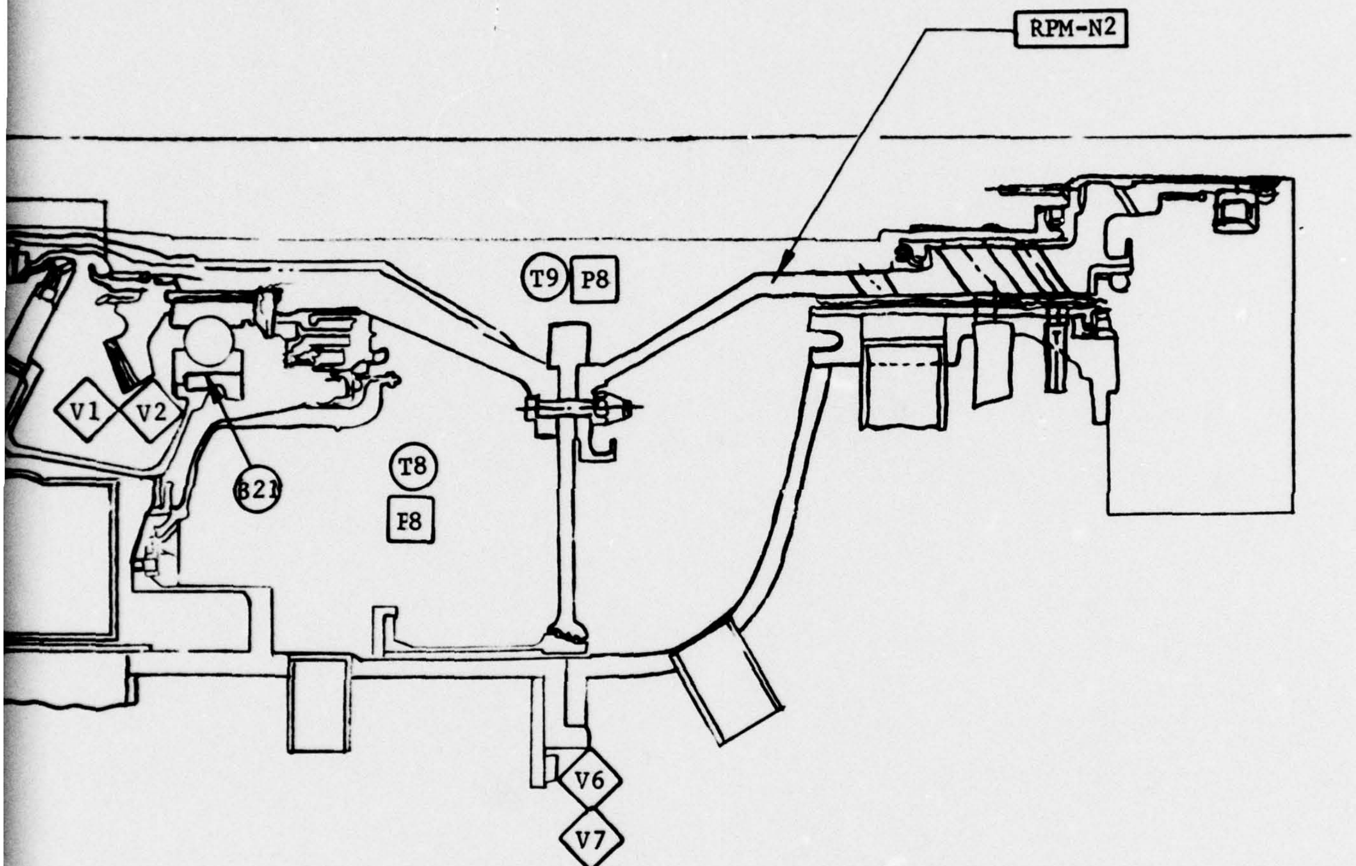


Figure 44. Compartmental Lubrication System Rig Instrumentation Schematic



stem Rig Instrumentation Schematic Longitudinal Cross Section

2

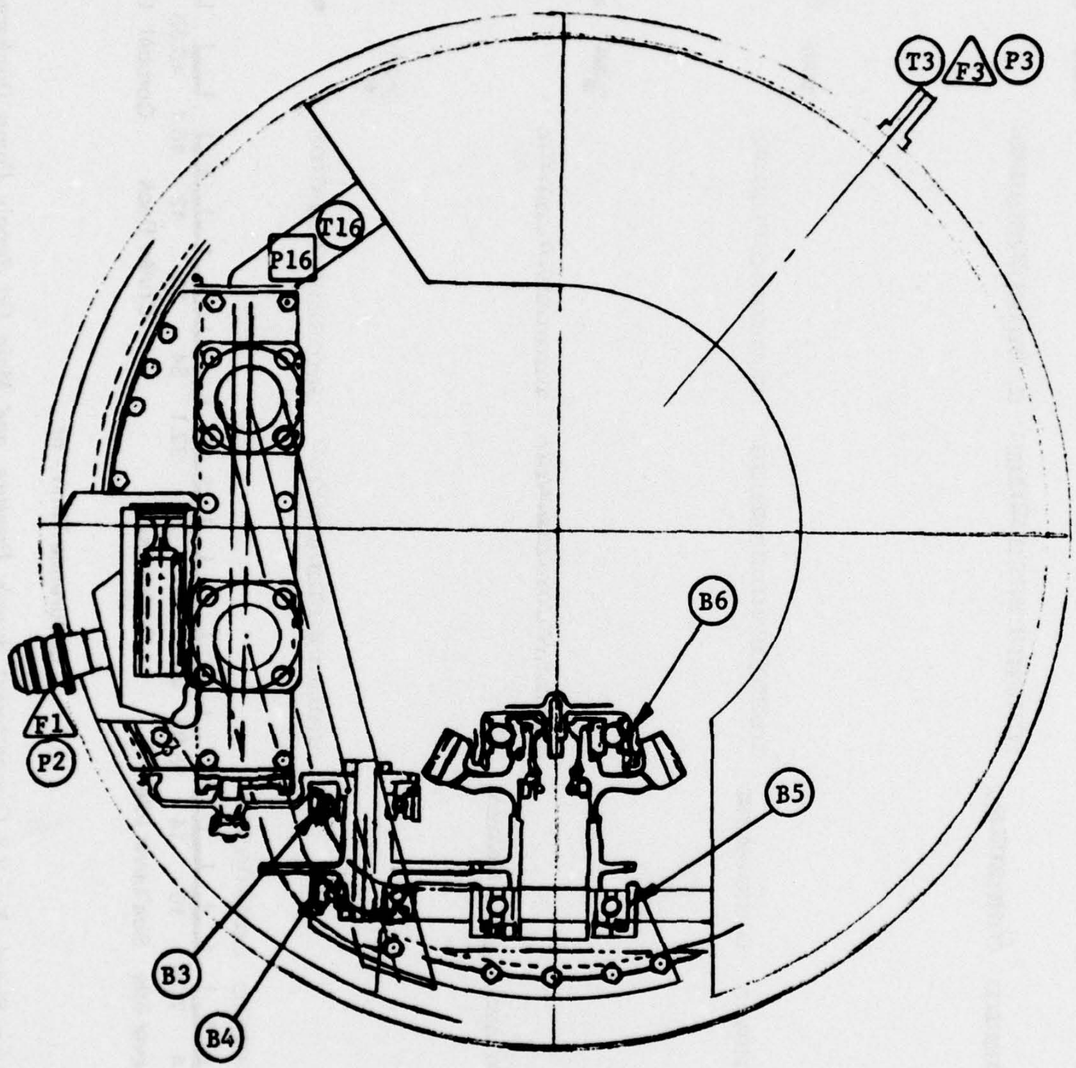


Figure 45. Compartmental Lubrication System Rig Schematic Traverse Cross Section

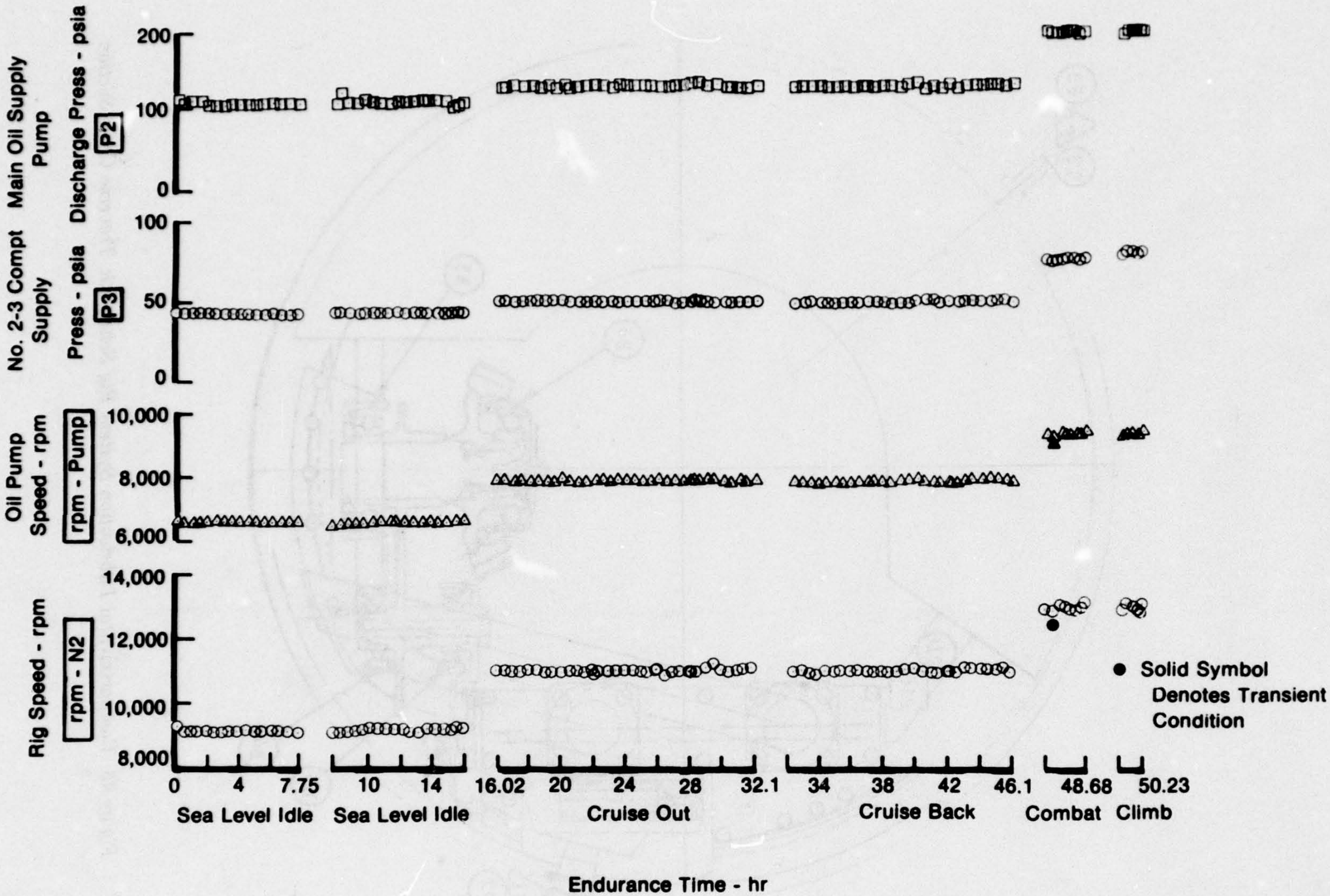


Figure 46. Oil Pump Speed, No. 2-3 Compartment Supply Pressure, and Main Oil Supply Pump Discharge Pressure vs Endurance Time

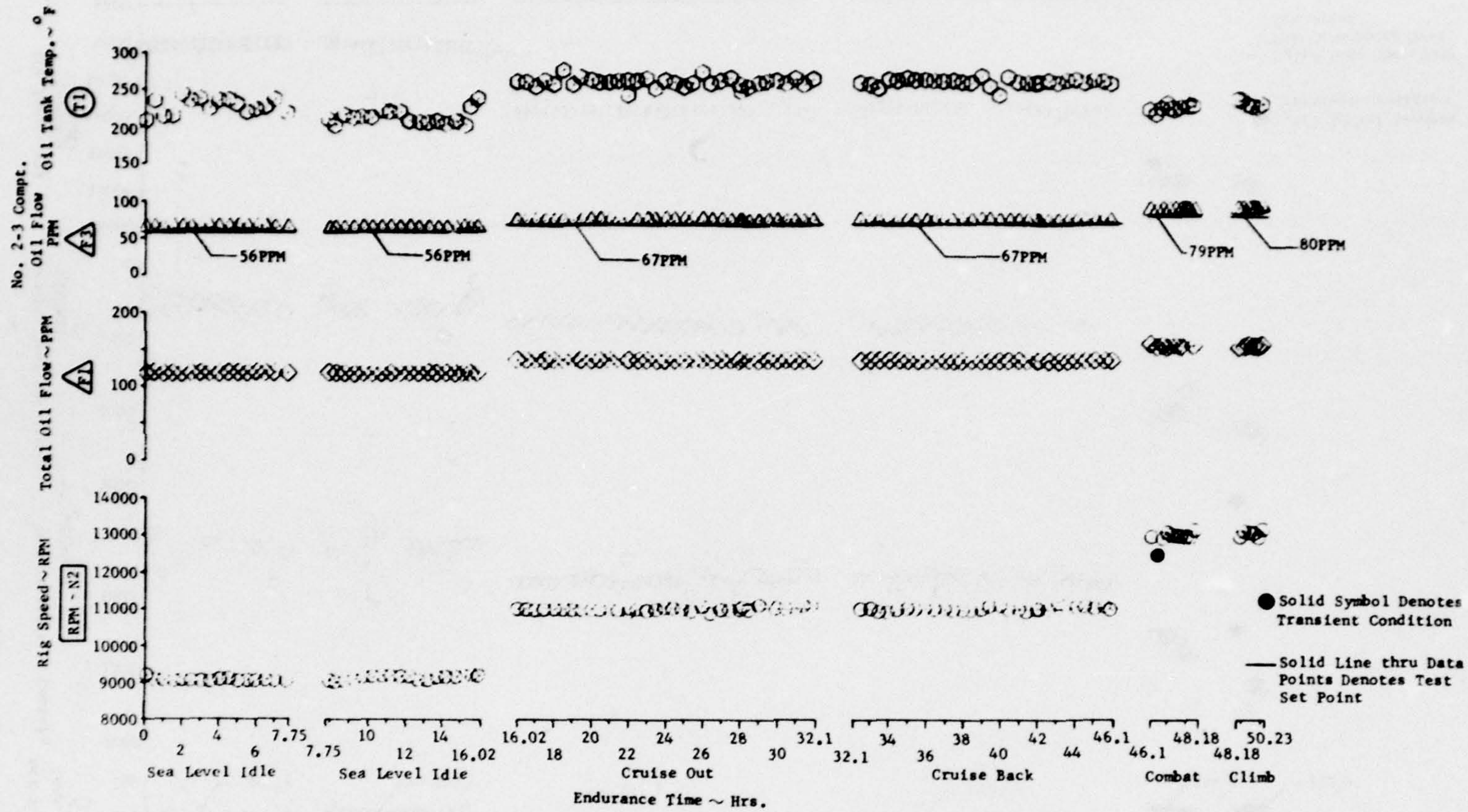


Figure 47. Total Oil Flow, No. 2-3 Compartment Oil Flow, and Oil Tank Temperature vs Endurance Time

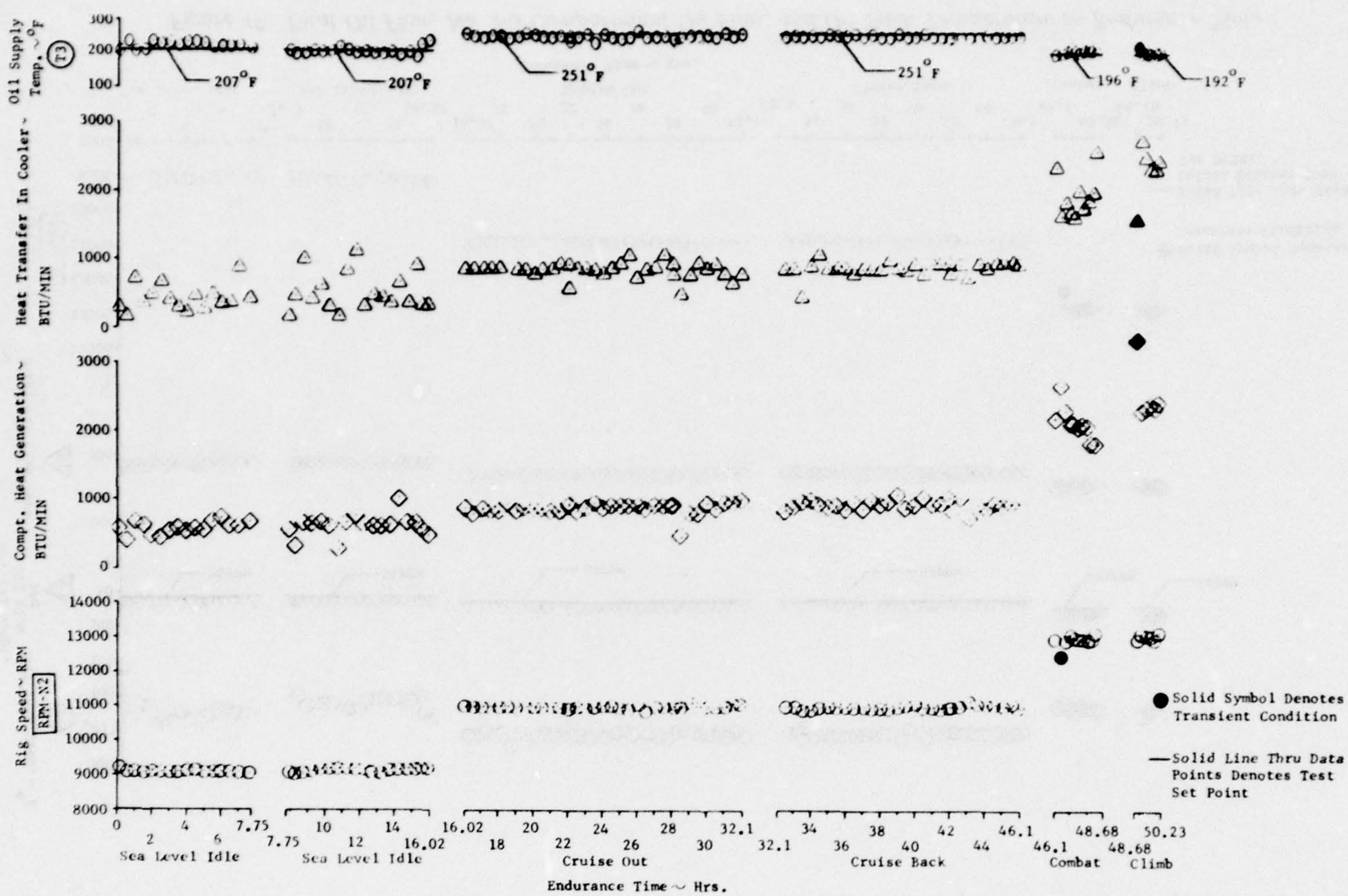


Figure 48. Compartment Heat Generation, Heat Transfer in Cooler, and Oil Supply Temperature vs Endurance Time

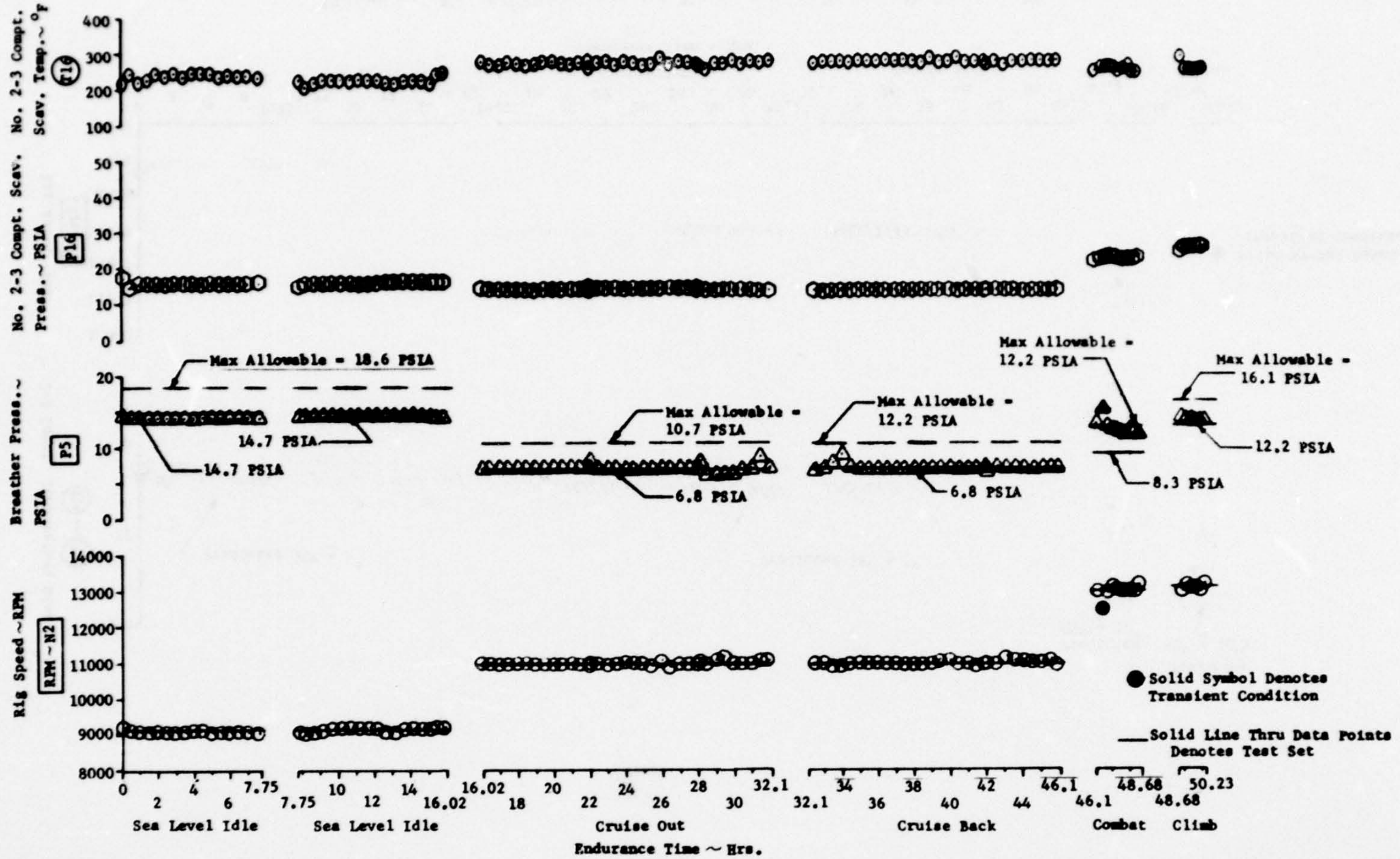


Figure 49. Breather Pressure, No. 2-3 Compartment Scavenge Pressure, and Temperature vs Endurance Time

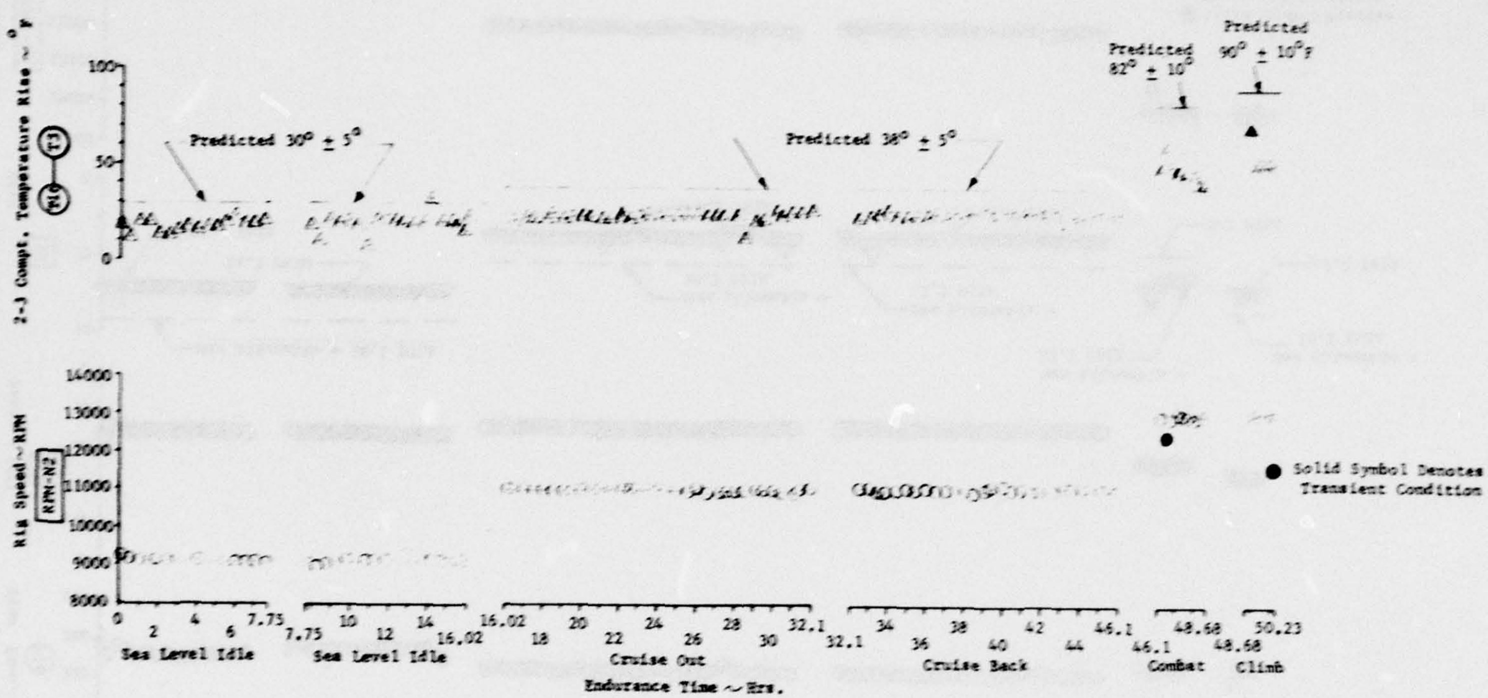


Figure 50. No. 2-3 Compartment Temperature Rise vs Endurance Time

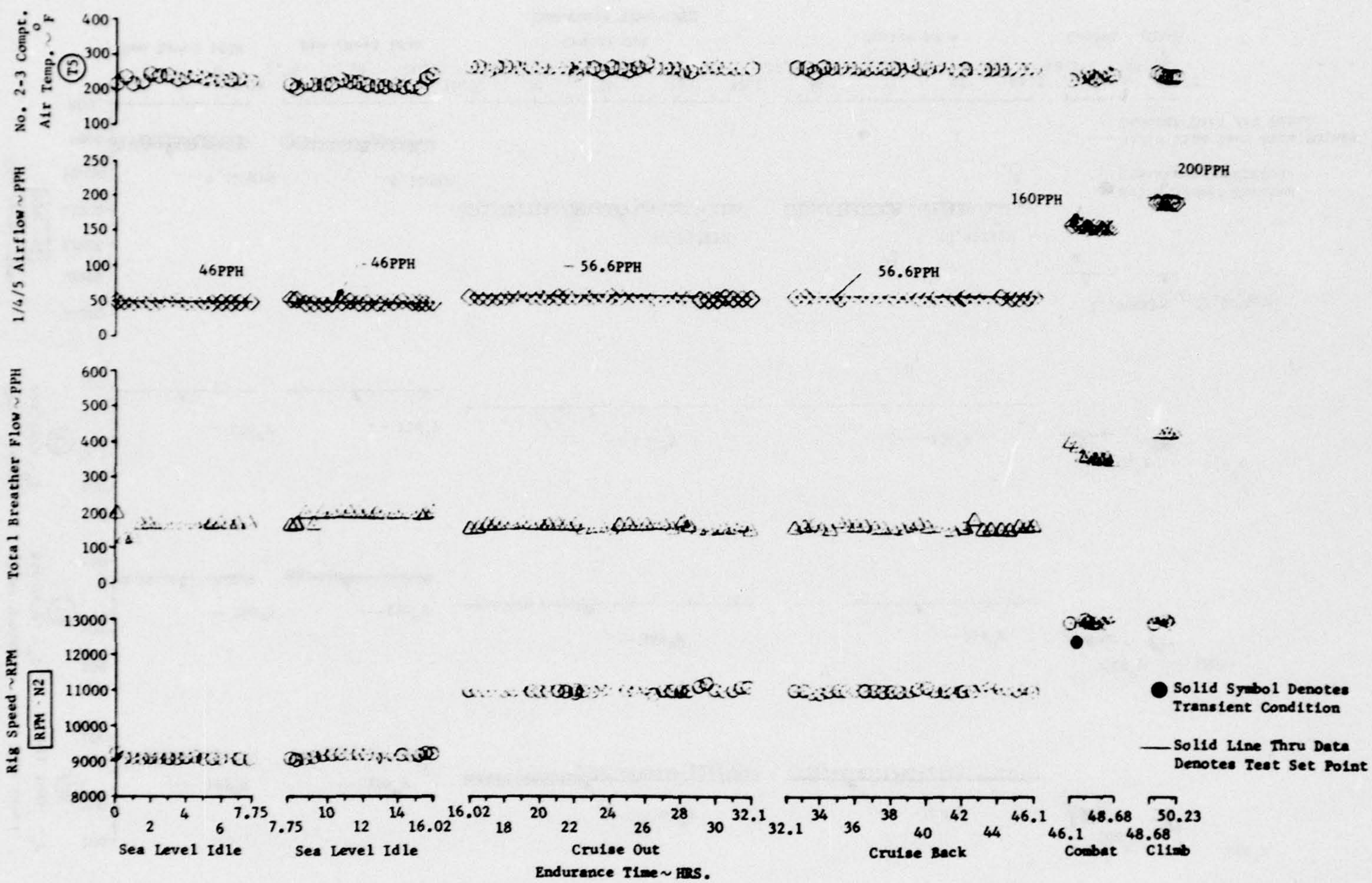


Figure 51. Total Breather Flow, No. 1, 4, and 5 Compartment Airflow, and No. 2-3 Compartment Air Temperature vs Endurance Time

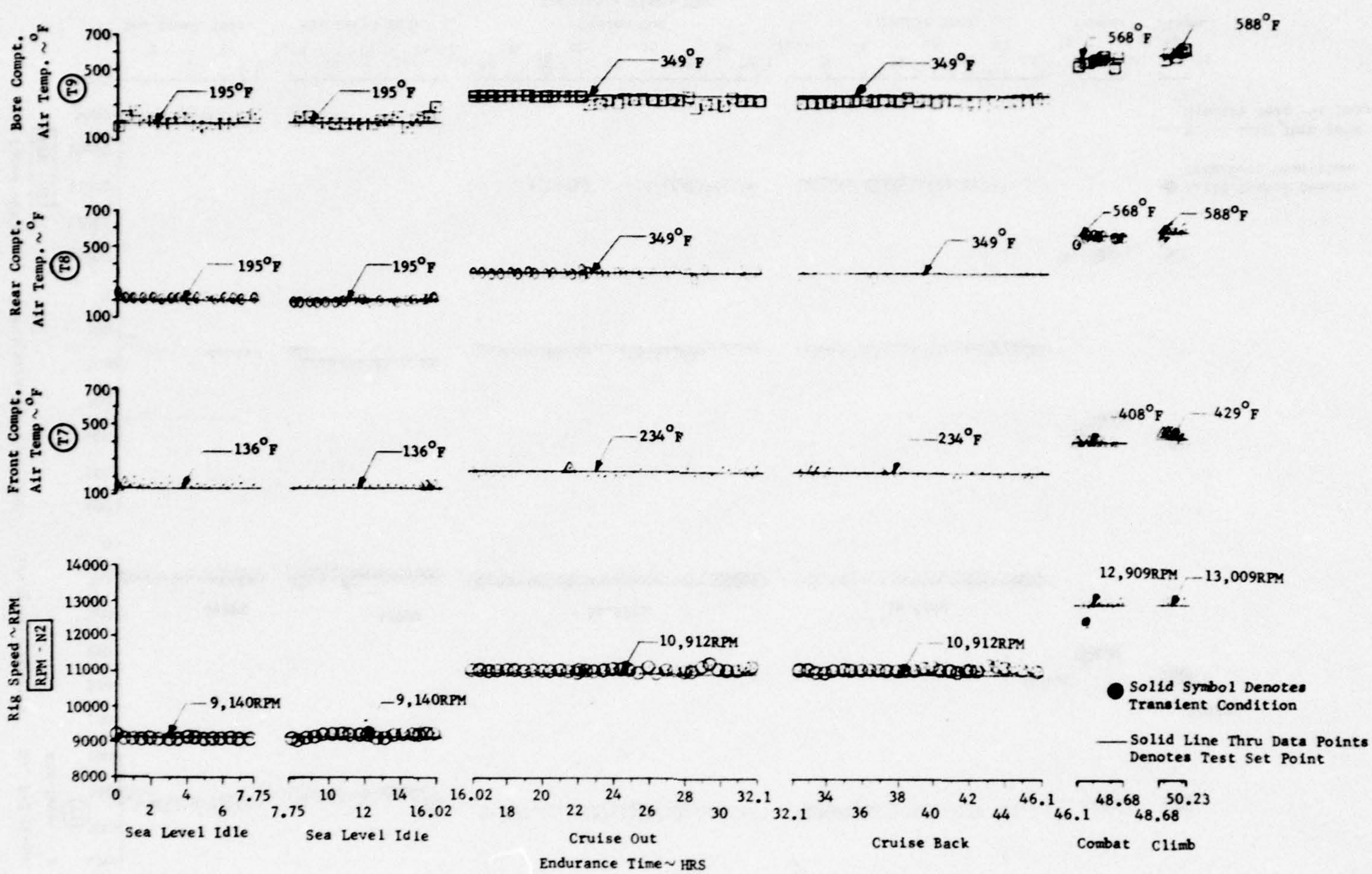


Figure 52. Front, Rear, and Bore Compartment Air Temperature vs Endurance Time

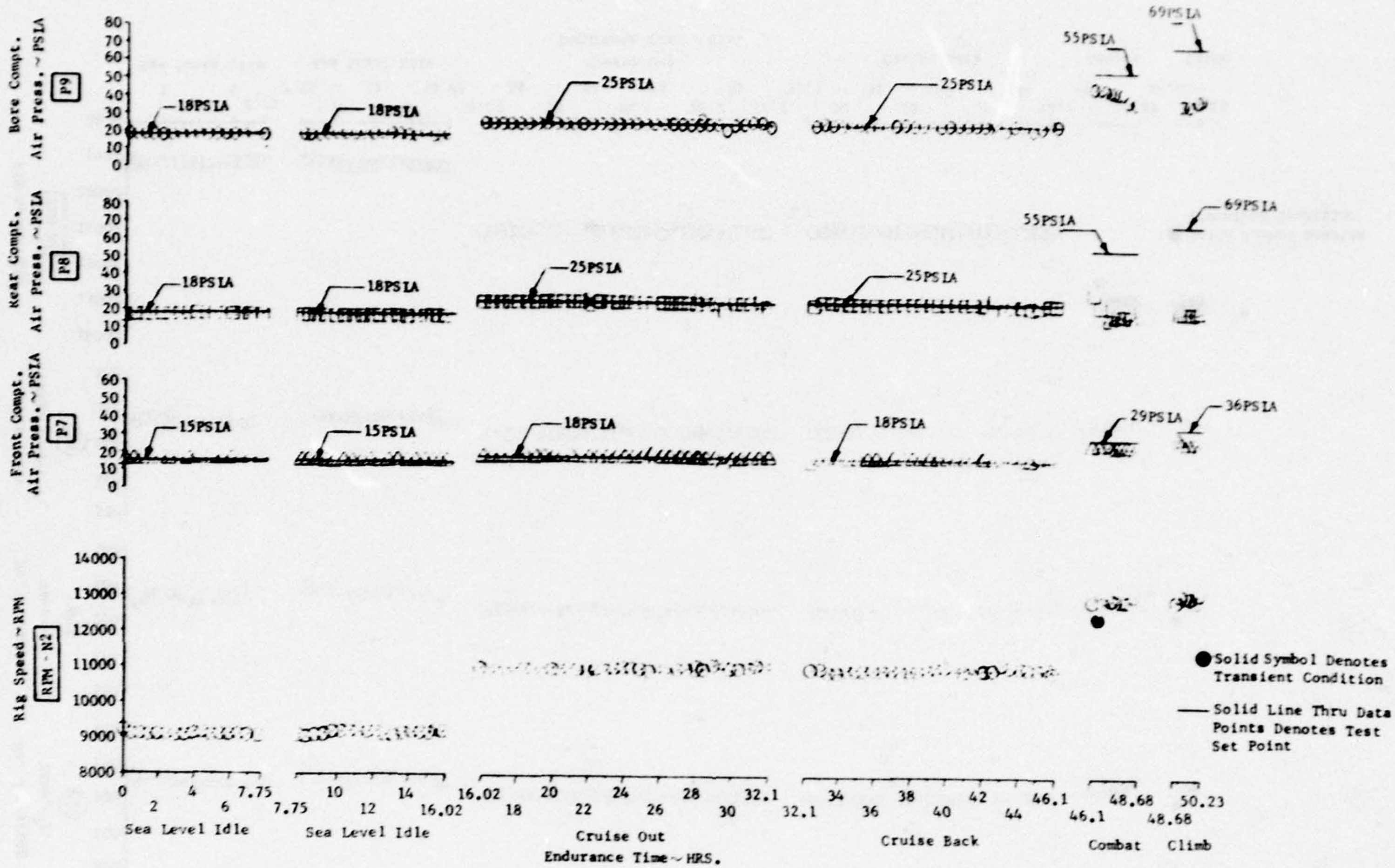


Figure 53. Front, Rear, and Bore Compartment Pressure vs Endurance Time

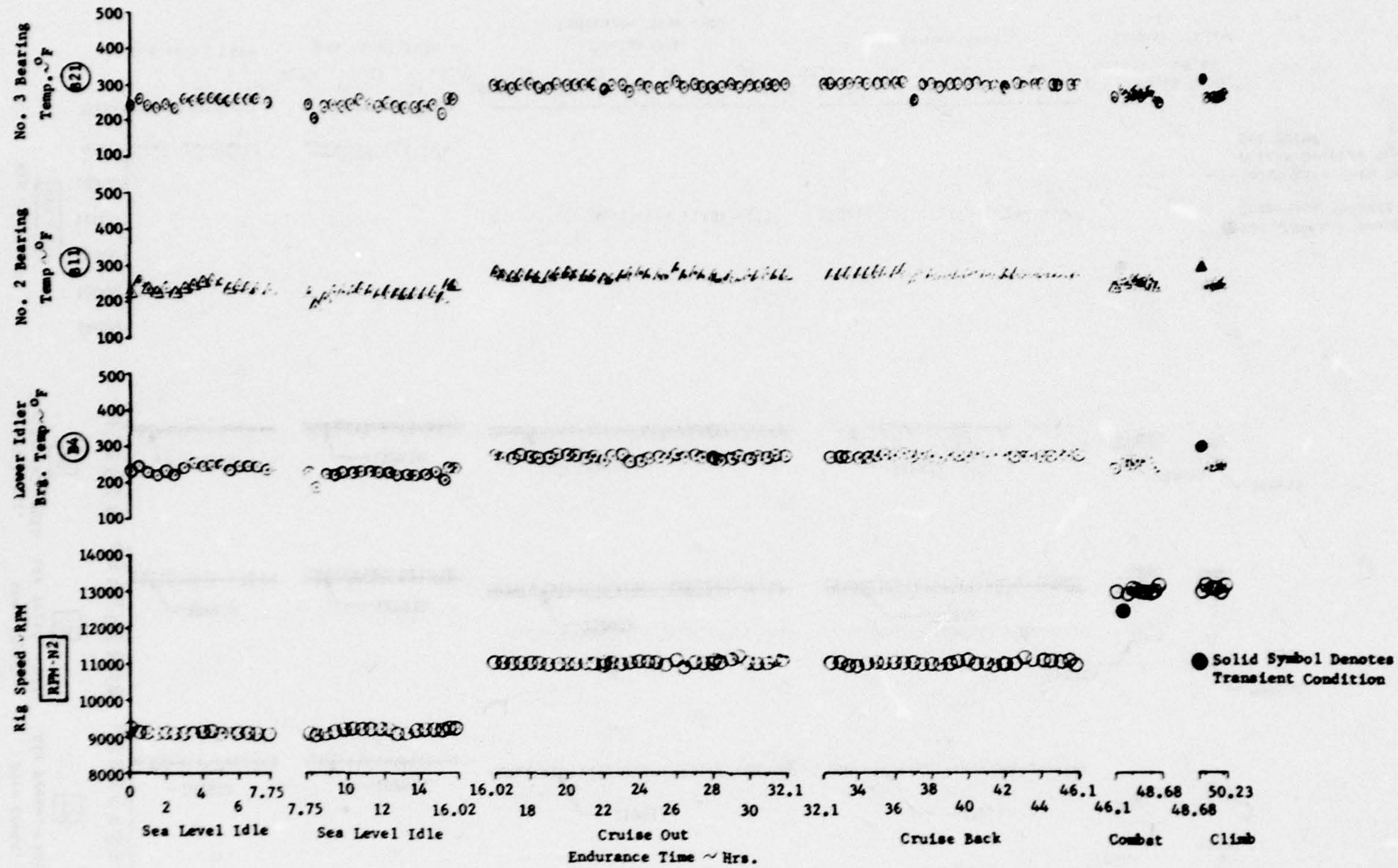


Figure 54. Low Idler, No. 2, and No. 3 Bearing Outer Race Temperature vs Endurance Time

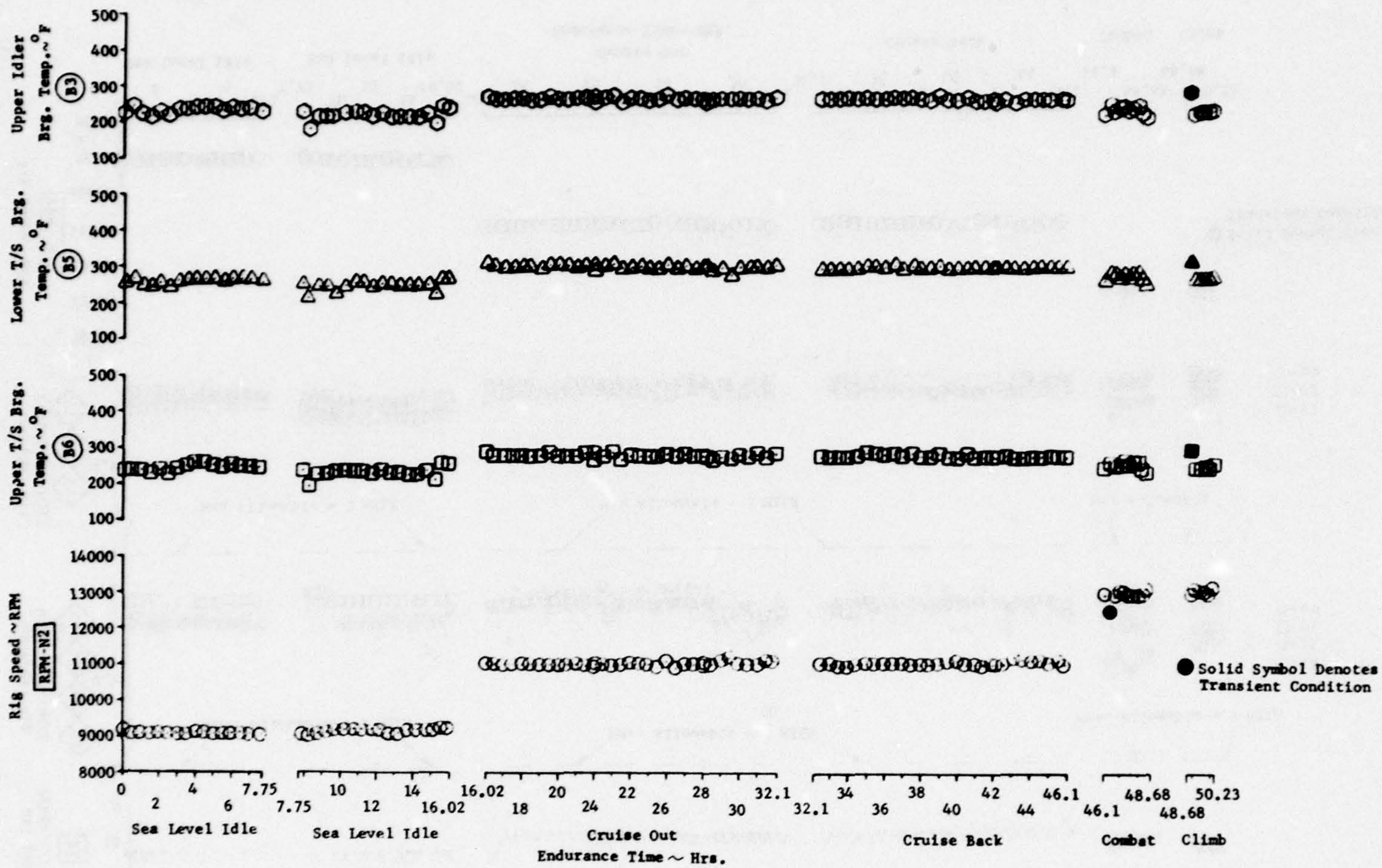


Figure 55. Upper Towershaft, Lower Towershaft, and Upper Idler Bearing Temperature vs Endurance Time

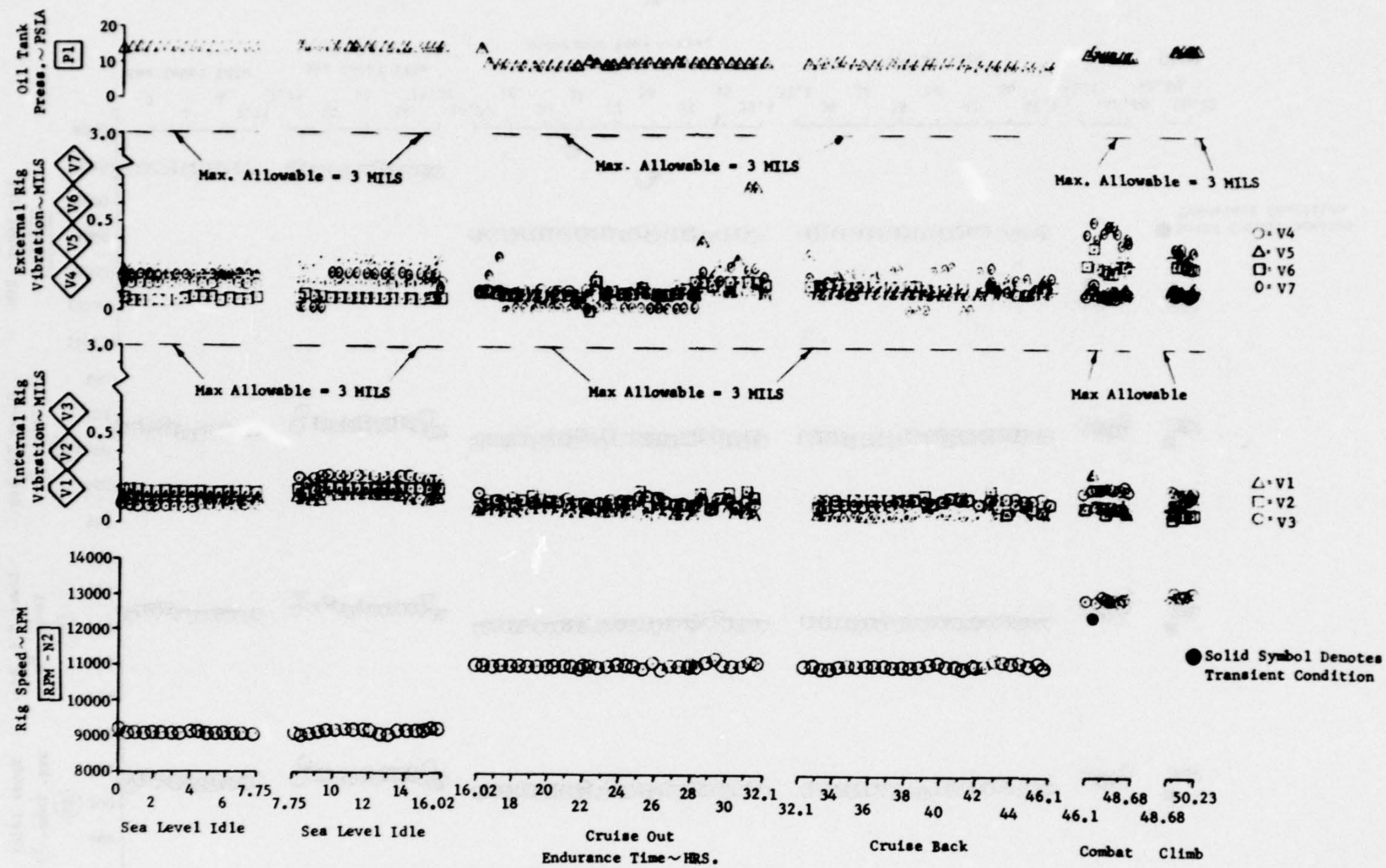


Figure 56. Internal and External Rig Vibration and Oil Tank Pressure vs Endurance Time

Figure 51 shows total breather air flow and simulated Nos. 1, 4, and 5 compartments air flow. Nos. 1, 4, and 5 compartment air flows simulated labyrinth real leakage from three compartments for the selected engine scheme. The difference between the total breather flow and simulated 1/4/5 compartment flow was No. 2/3 compartment leakage.

The rig has F100-PW-100 Bill-of-Material carbon seals which leak about 25 pph. The No. 2/3 compartment leakage was 10 times higher than expected for carbon seals. The leakage from the rear compartment into the No. 2/3 compartment caused high breather flow and entrainment of compartment oil out the breather.

Figures 52 and 53 show environmental temperatures and pressures surrounding the No. 2/3 compartment, respectively. The front, rear, and bore compartment (P7, P8, and P9, respectively) pressures were set lower than test point conditions during the combat and climb mission points in order to limit leakage into the compartment. It was felt that breather pressure was a parameter that directly affected the test articles more than environmental pressures surrounding the No. 2/3 compartment. Therefore, the highest possible environmental pressures were set based on maximum allowable breather pressure (P5), i.e., 8 inches Hg over test set point.

Figures 54 and 55 show high-speed gear train bearing outer race temperatures and rig No. 2 and No. 3 bearing outer race temperatures. There was no abnormal temperature rise indicated. The lower tower shaft bearing showed the highest operating temperature of 300°F at cruise conditions and a maximum temperature rise over oil supply temperature of 83°F at climb conditions.

Rig internal and external vibrations are shown in Figure 56. A maximum allowable limit of 3.0 mils vibration was selected. At no time during the endurance run did any vibration level exceed 1.0 mil with internal vibrations consistently below 0.3 mil.

b. SOAP and Chip Detector Analysis Results

Oil samples were taken at 2.75, 6.75, 16.0, 20.0, 22.0, 28.0, 42.0, 46.0 and 50.23 hours during the 50-hour endurance test. Iron content varied throughout the 50 hours and was the element found most abundant in the oil. Iron content ranged from less than 1.0 to as high as 6.4 parts per million. Traces of aluminum, nickel, silver, chromium, and titanium were found (less than 1.0 part per million) and continued at those low levels throughout the test. Initial samples taken showed slightly higher aluminum (3.2 ppm) and can be attributed to pump wear-in.

Analysis of material collected by the rig magnetic chip detector showed iron/nickel, chromium, and aluminum. Again, early samples showed higher iron/nickel content due to gear train wear-in and flushing of rig interior

Analysis of filter bowl residue showed traces of carbon. The percentage amount increased slightly throughout the test showing some carbon seal wear.

Early in the endurance test, three large metal particles were found in the filter bowl. Particles were approximately 0.150 × 0.100 inch and resembled instrumentation tack straps. Analysis confirmed that the material was Inconel 600 shim stock used to secure instrumentation leads on the rig interior.

c. Disaster Monitoring O'Graph

At all times, when the endurance test was in progress, rig speed and selected temperatures, pressures, and vibrations were monitored and recorded on light sensitive o'graph paper. The data were not reduced and served only for investigative purposes in cases of rig malfunction.

d. System Rig Post-Run Teardown Results

Upon completion of the 50-hour endurance test the rig was dismounted from the stand for disassembly. There was no evidence of coking on any internal rig parts.

Figure 57 shows the F100-PW-100 No. 3 rear carbon seal support and spiral wound gasket. The excessive breather flow was caused by leakage of rear chamber air through the No. 3 rear carbon seal support and the rig main housing mount flange. The leakage was caused by an improperly installed spiral wound crush gasket. The installation of the gasket and seal support is a blind assembly in the rig. The gasket damaged area is shown in Figure 58.

Figure 59 shows the high-speed gear train as removed from the F100-PW-100 No. 2/3 crossover housing. Very slight wear patterns were noted on the towershaft spur gear. Gear tooth wear was negligible on all gears.

A disassembled view of the high-speed gear train is shown in Figure 60.

The lower towershaft bearing showed a slight discoloration of the split inner race and is shown in Figure 61. At 100X power, (Figure 62) the surface texture of the balls shows the results of small particle contamination damage. The lower towershaft bearing is the lowest point in the compartment. Any foreign particles in the compartment would be flushed down to the area of the lower towershaft bearing.

The high-speed oil supply and scavenge pump showed a reddish discoloration on all internal and external surfaces that were in direct contact with the oil. This discoloration was only present on the three anodized aluminum housings and end plate. Fabrication Research personnel indicated that the synthetic engine oil, MIL-L-7808G reacted with the anodized surfaces causing the surface to have a stained appearance.

Figures 63 and 64 show the supply pump gears, and Figure 65 shows the scavenge gears. All are from high-speed oil supply and scavenge pumps S/N 1. Total time on this pump is 87 hours. The figures show there is a slight discoloration on the ends of each journal of both supply pump packages. This is due to contact with the rubber lip seals used on the supply pump packages. A 20X photo of the rubber lip seal is shown in Figure 66. After 87 hours run time all lip seals in the supply pump showed signs of considerable wear. This is an area which will have to be investigated for future applications of a high-speed pump. On this application, where the pump is located inside the bearing compartment, a shaft seal oil leak would have only a slight effect on pump performance and would not result in external engine oil leakage.

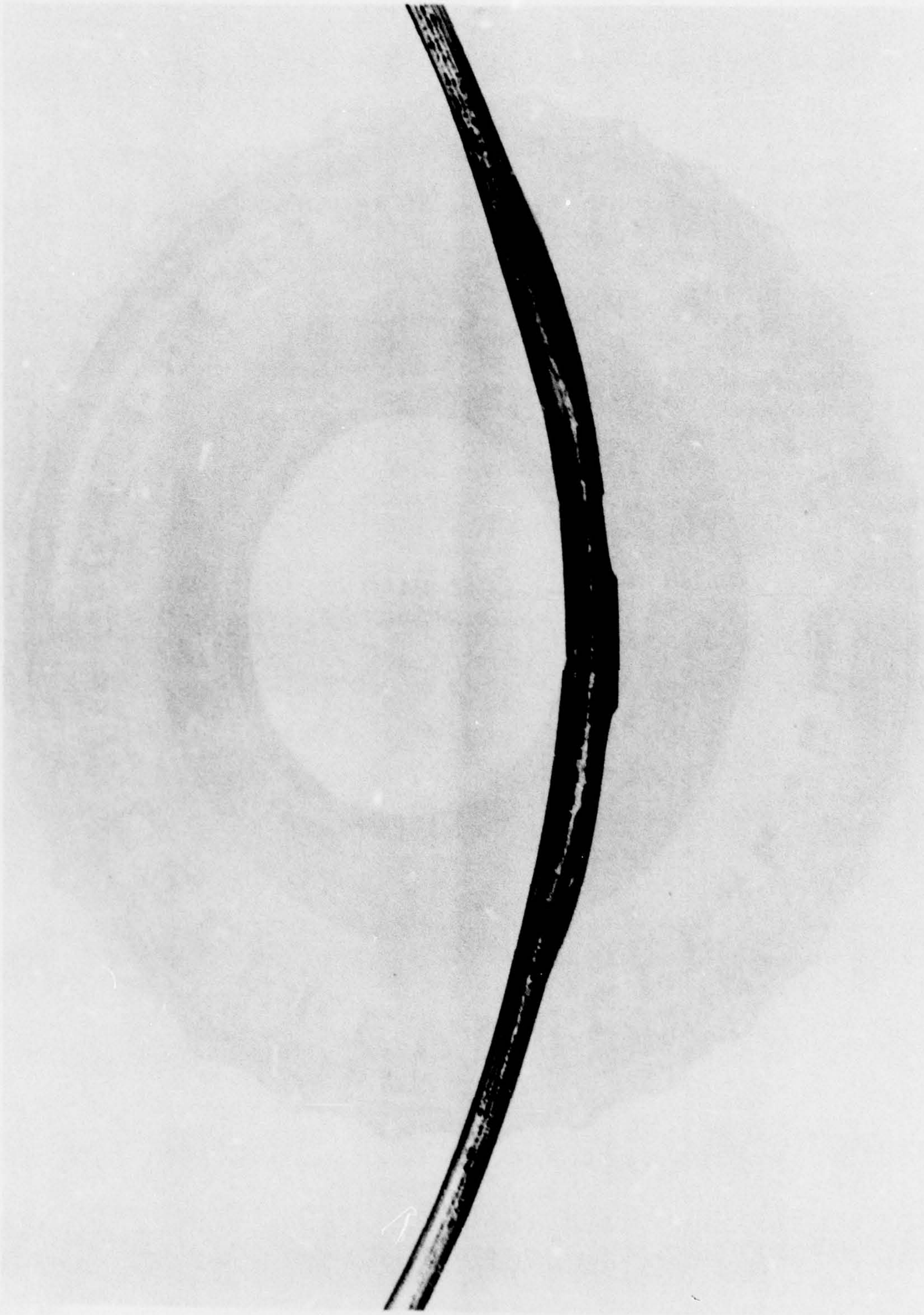
The gear teeth on both the supply and scavenge packages showed no abnormal wear. The backlash of each package is shown in Table 28.

TABLE 28. PUMP GEAR TEETH BACKLASH

Run Time (hr)	*Drive End Supply Package Backlash (in.)*	*Supply Package No. 2 Backlash (in.)*	*Scavenge Package Backlash (in)*
0	0.0045	0.0045	0.0045
20	0.0068	0.0055	0.0058
87	0.0068	0.0055	0.0058

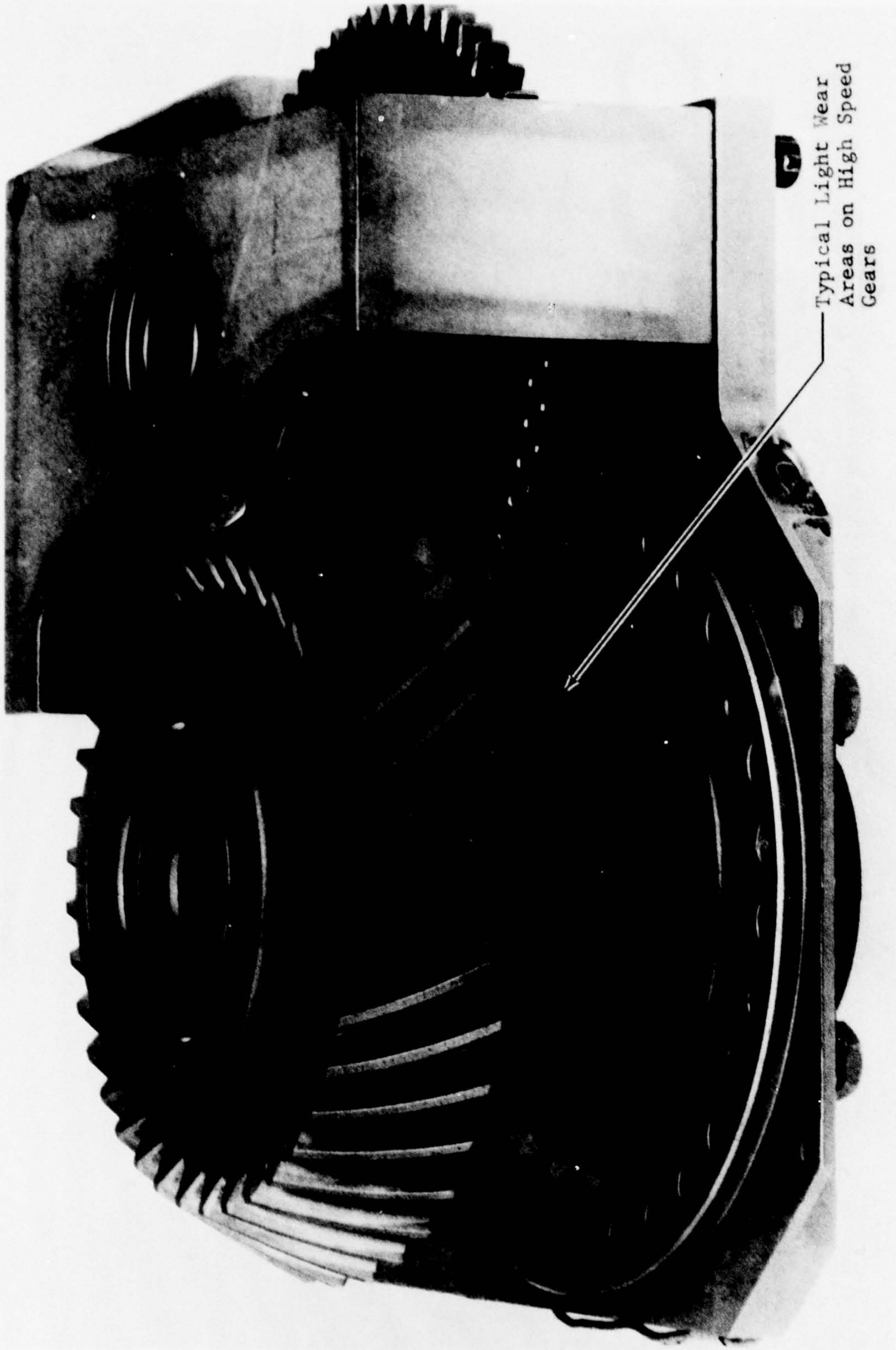
FAE 165520

Figure 57. F100 No. 3 Compartment Rear Seal Support



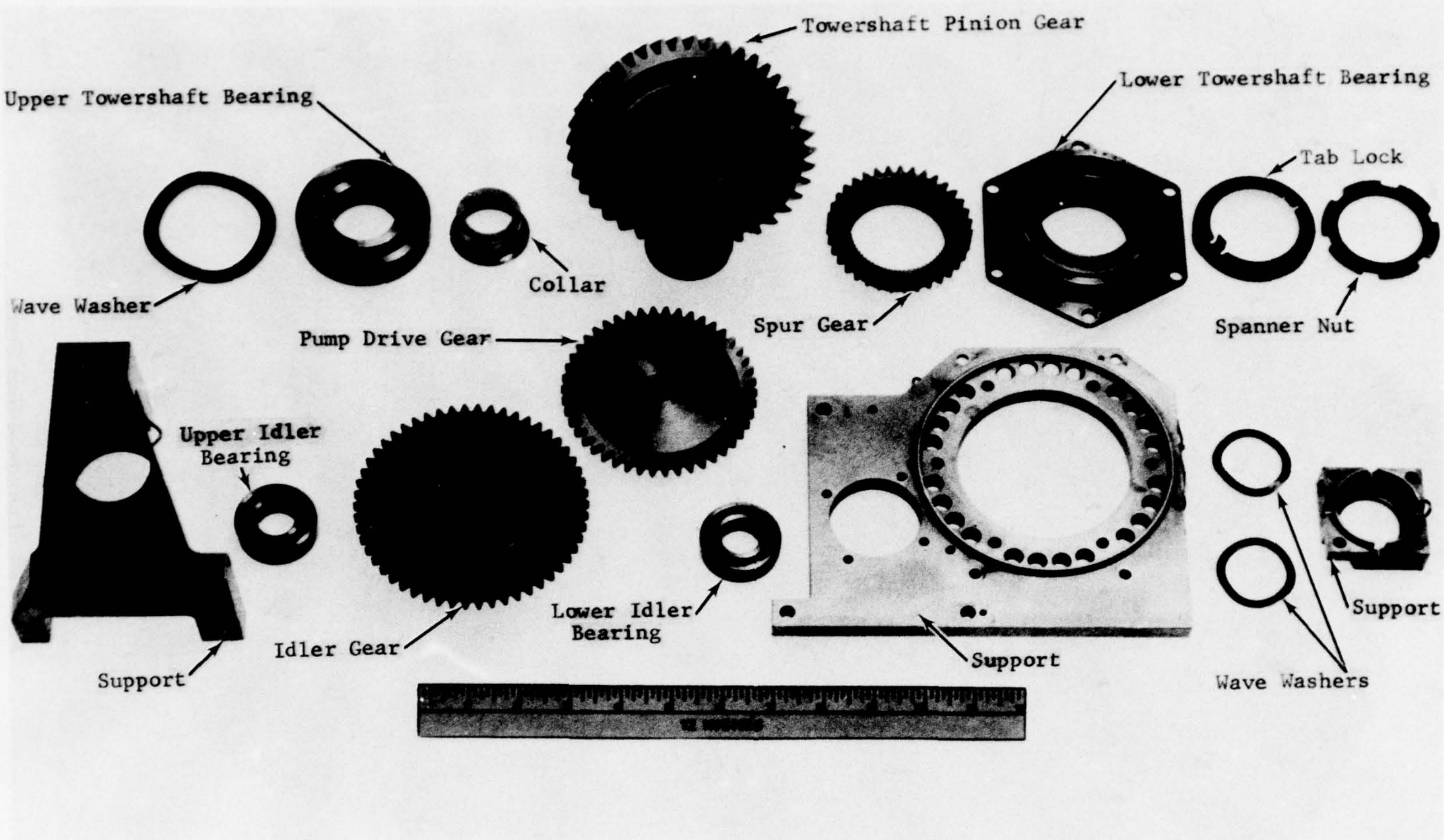
FAE 165523

Figure 58. Damaged Spiral Wound Gasket



FE 165587

Figure 59. High-Speed Gear Train



FE 166074

Figure 60. Disassembled View of High-Speed Gear Train

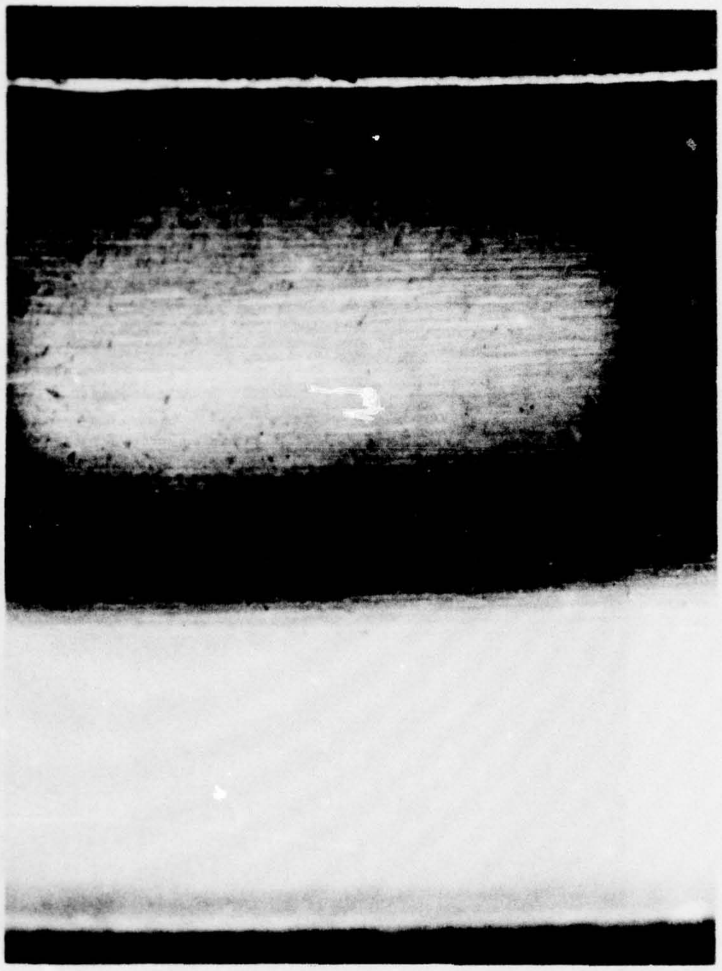
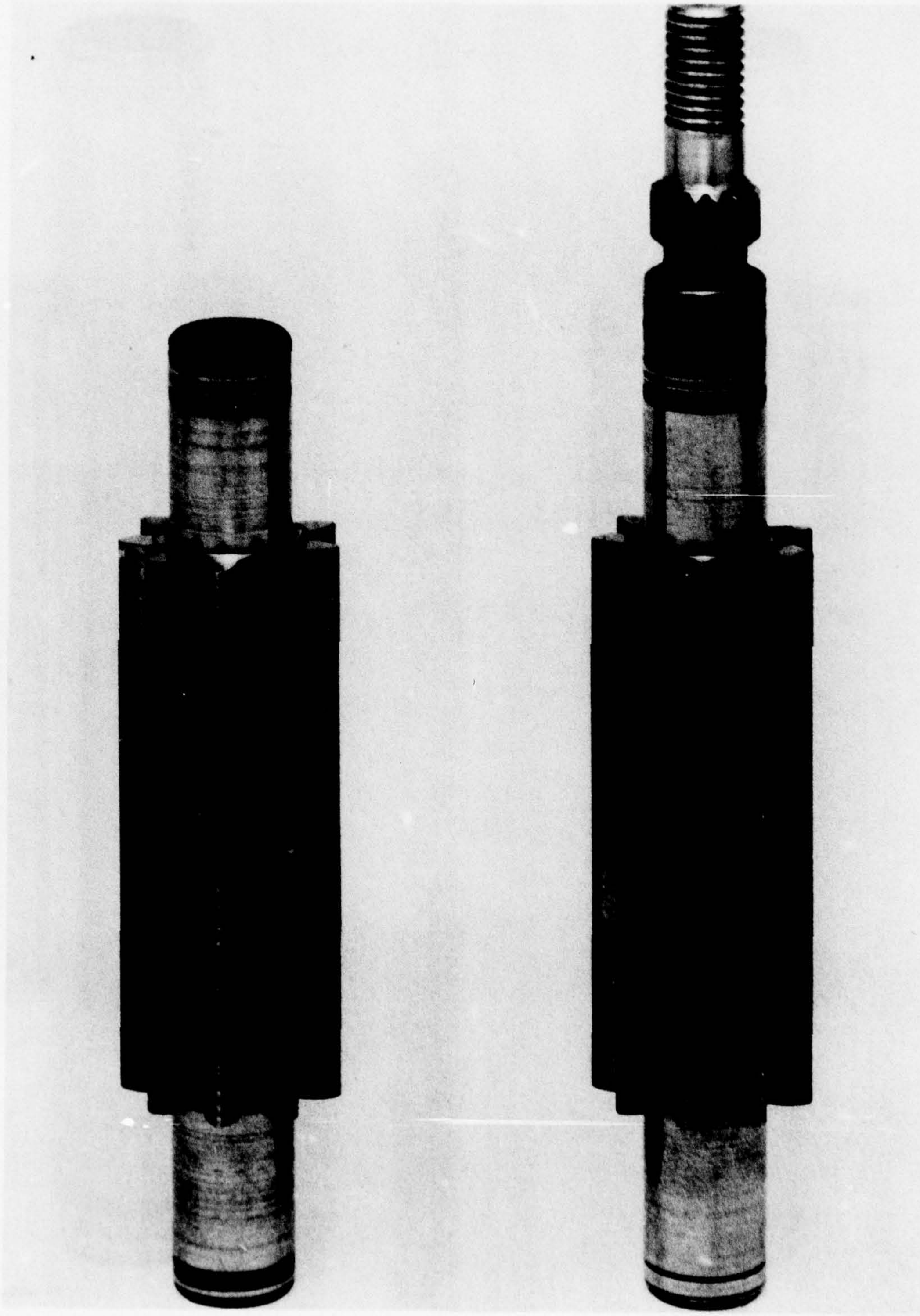


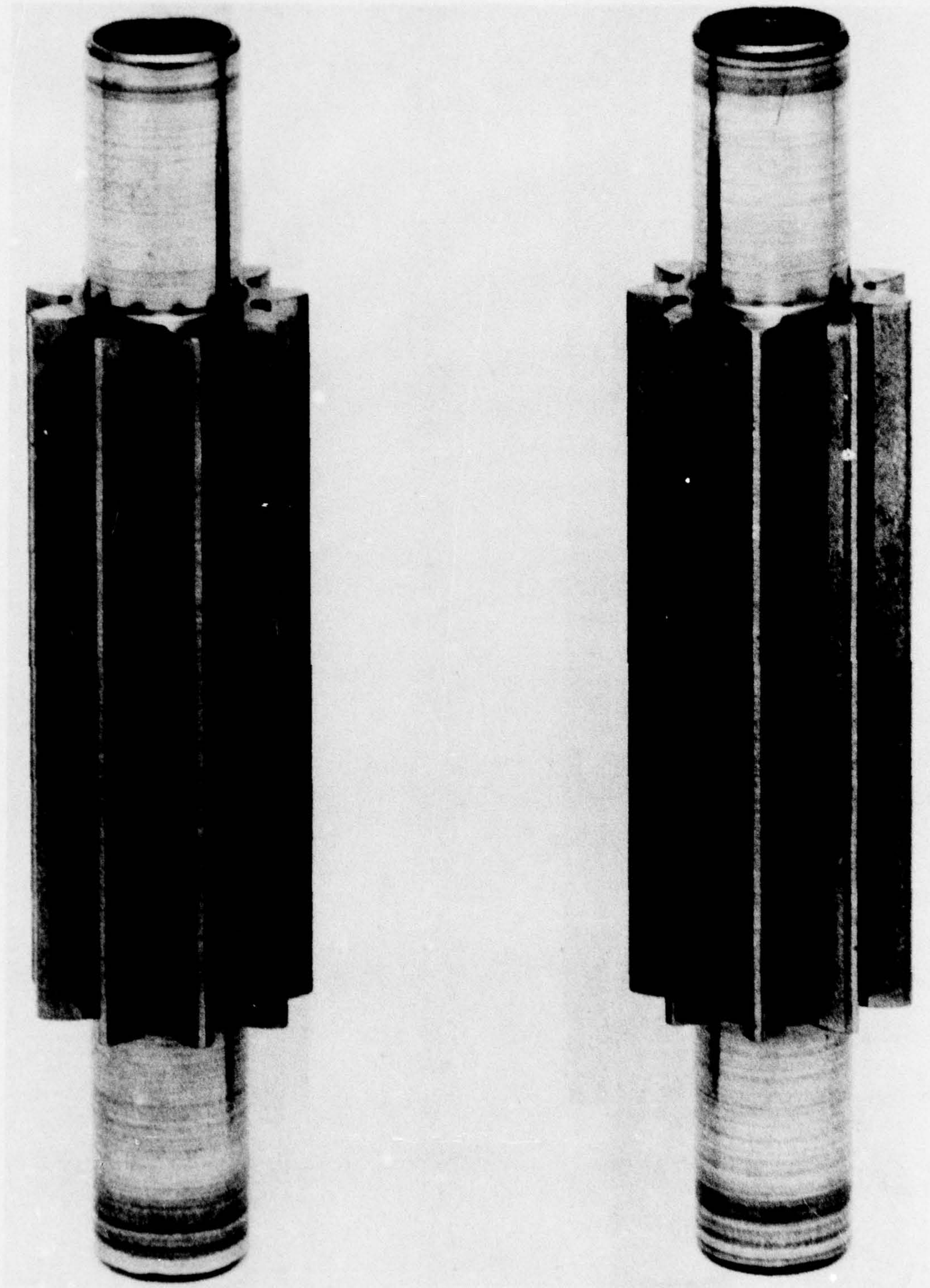
Figure 61. Lower Towershaft Bearing Inner Race Contamination Damage Magnified 15 Times

Figure 62. Lower Towershaft Bearing Ball Contamination Damage Magnified 100 Times



FE 165557

Figure 63. Supply Pump Gearshafts, Drive End Supply Package



FE 165556

Figure 64. Supply Pump Gearshafts, Package No. 2

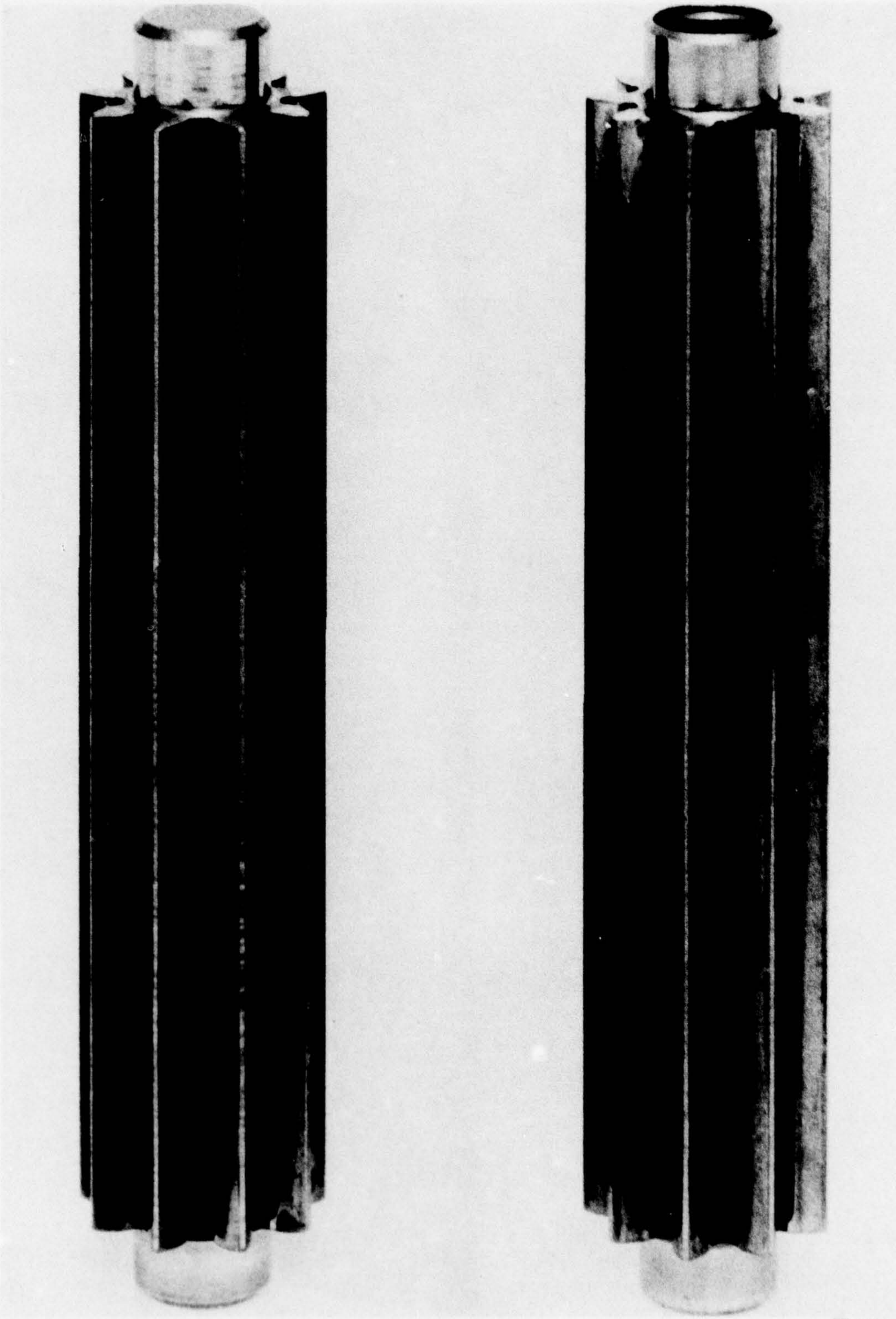
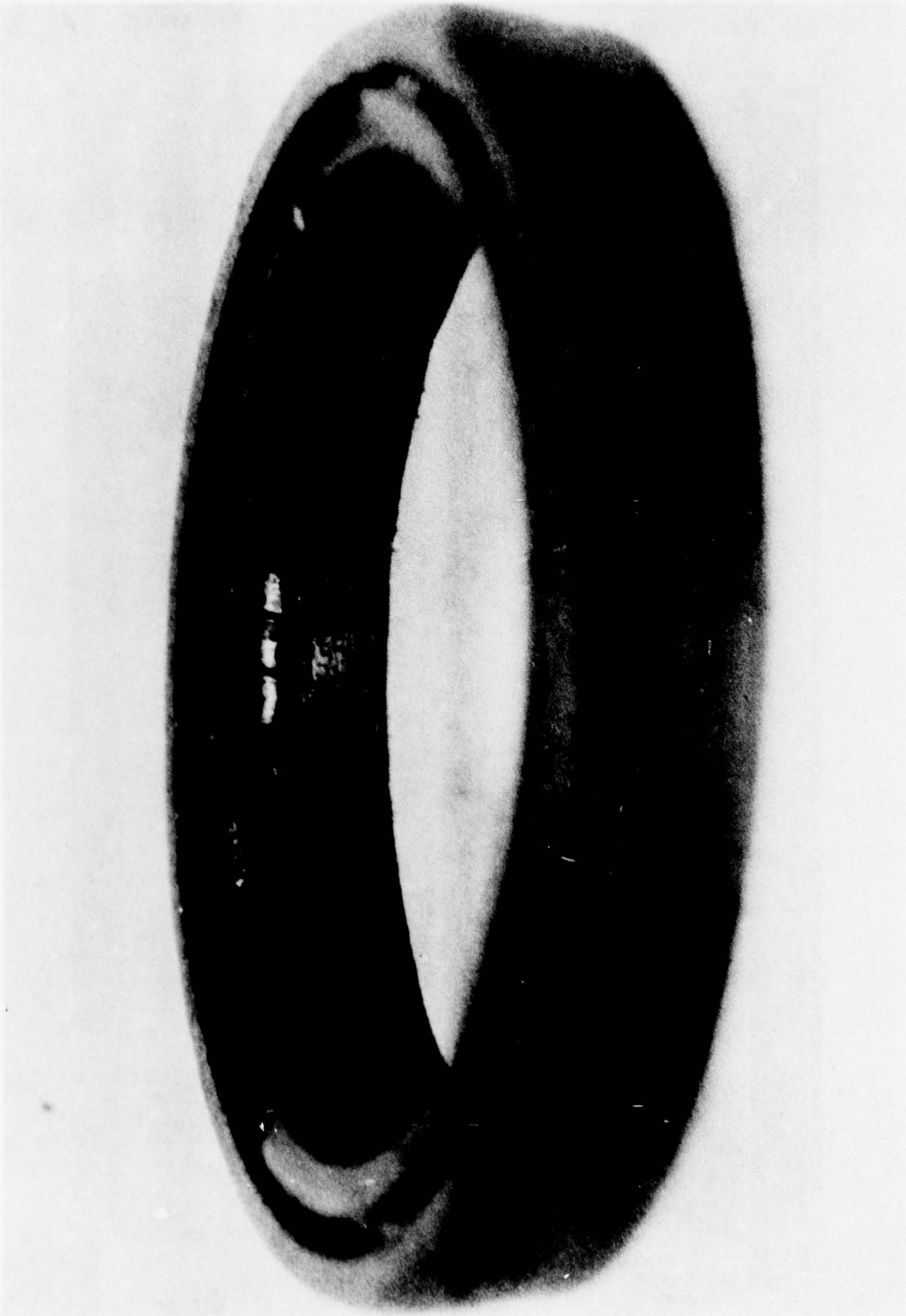

FE 165555

Figure 65. Scavenge Pump Gearshafts



FAL 46347

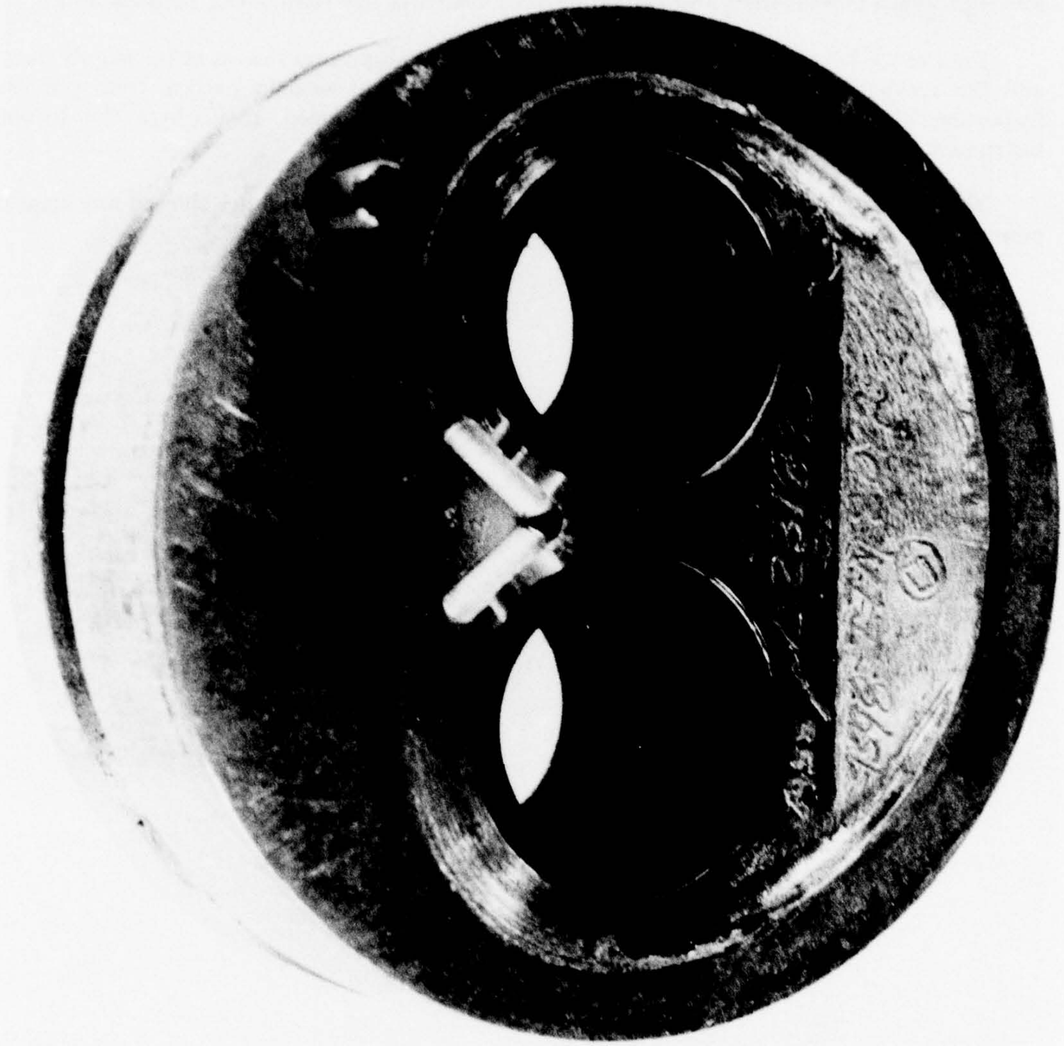
Figure 66. Worn Supply Pump Rudder Lip Seal Magnified 20 Times

Wear patterns on the journals of the supply and scavenge pump gears are due to small particle (less than 70 micron diameter) contamination. This wear pattern is more noticeable on the supply pump journals, Figures 63 and 64, than on the scavenge pump journals, Figure 65, since the supply journals are more heavily loaded.

Figures 67 through 72 show the pump journal bushings. Average supply pump shaft wear was 0.0001 inch. Average supply pump carbon journal bushing wear was 0.0002 inch. Average scavenge pump journal shaft and carbon bushing wear was less than 0.0001 inch for both.

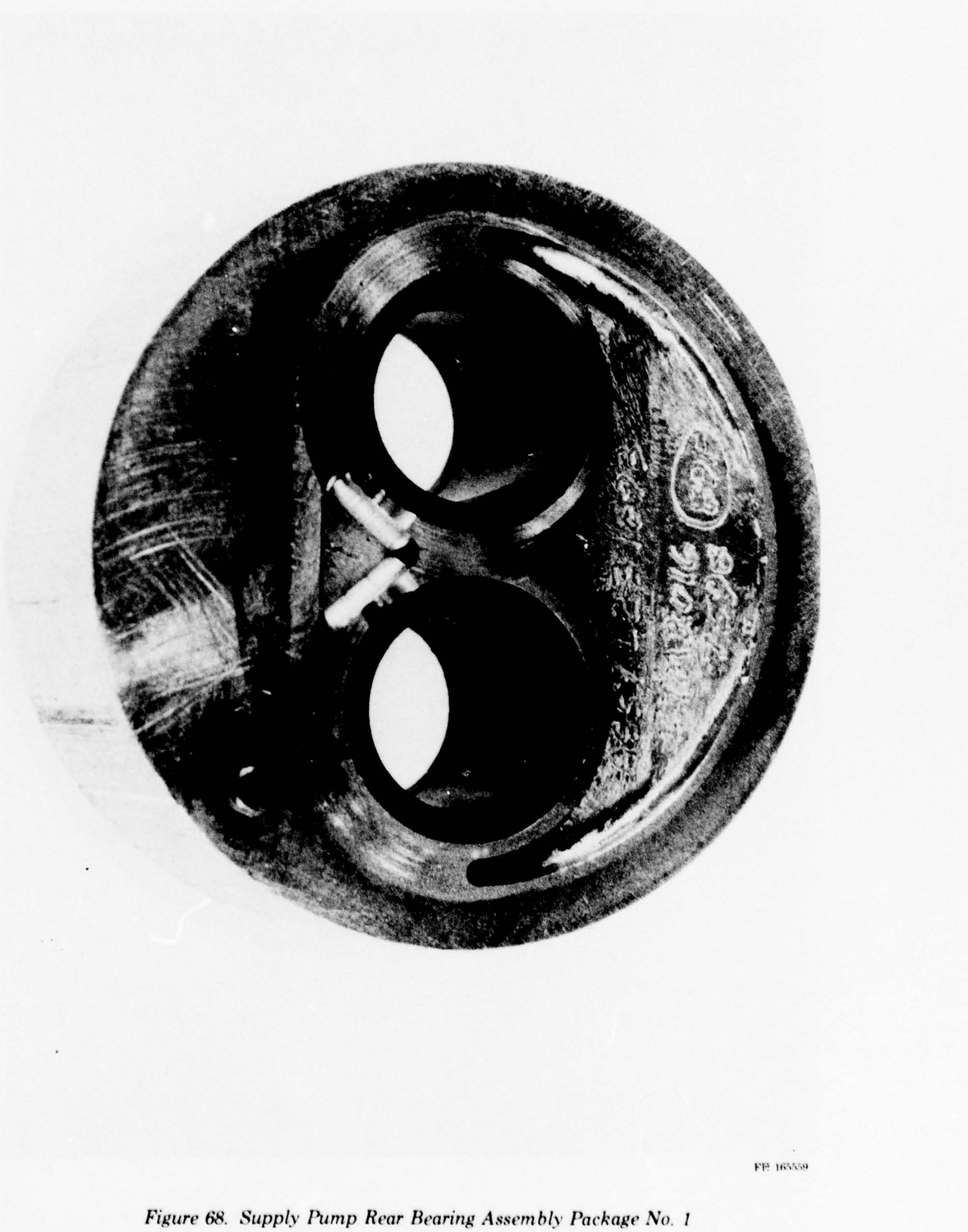
Figures 73, 74, and 75 show the wear-in areas of the aluminum sleeves of the supply pump and the scavenge pump. There is no noticeable change in wear-in pattern from previous inspection at 20-hours run time except for the local damaged area where the Inconel instrumentation tack strap was passed through.

Neither the aluminum sleeves nor the aluminum bearing assemblies showed any signs of pump cavitation damage.



FE 163558

Figure 67. Supply Pump Front Bearing Assembly Package No. 1



FE 165559

Figure 68. Supply Pump Rear Bearing Assembly Package No. 1

FE 165560

Figure 69. Supply Pump Front Bearing Assembly Package No. 2

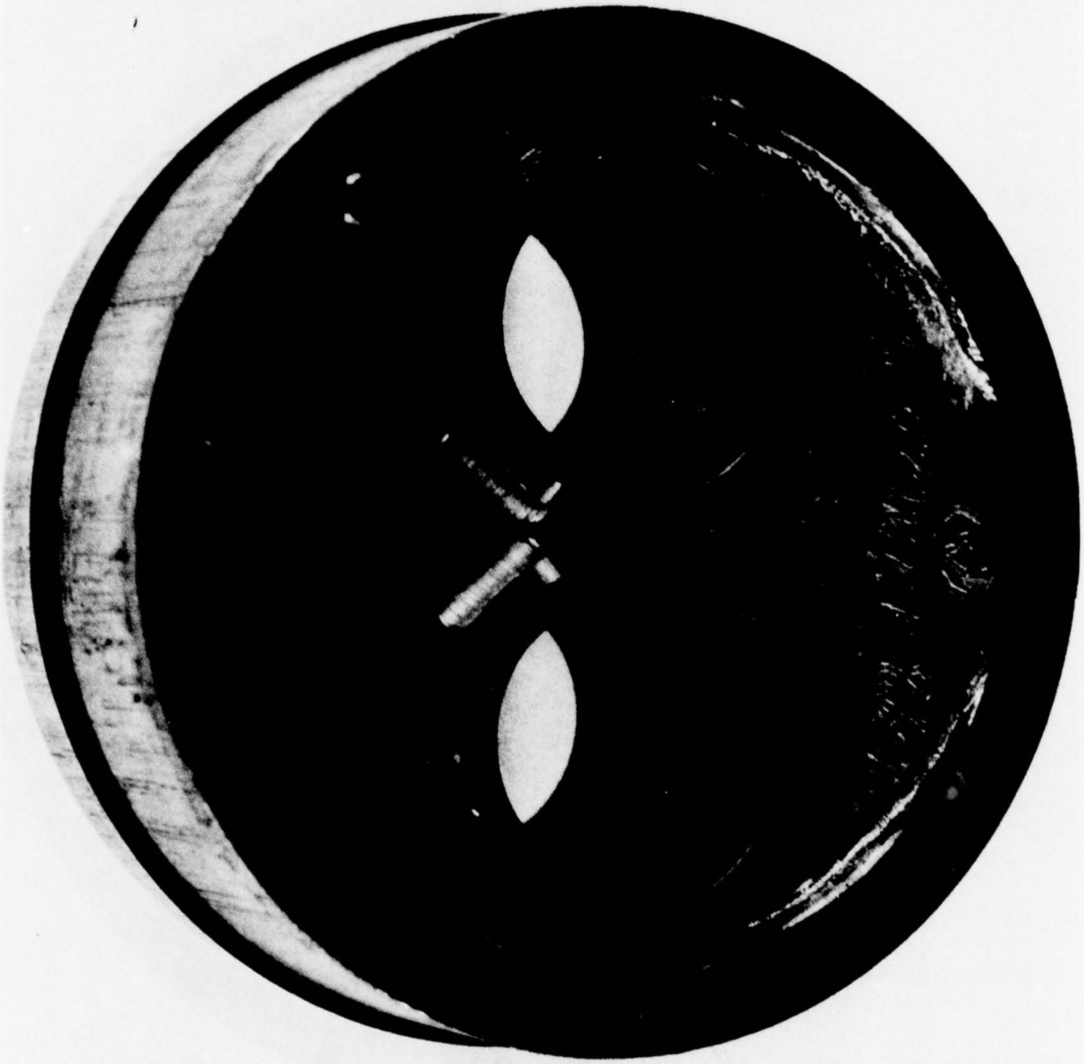


Figure 70. Supply Pump Rear Bearing Assembly Package No. 2

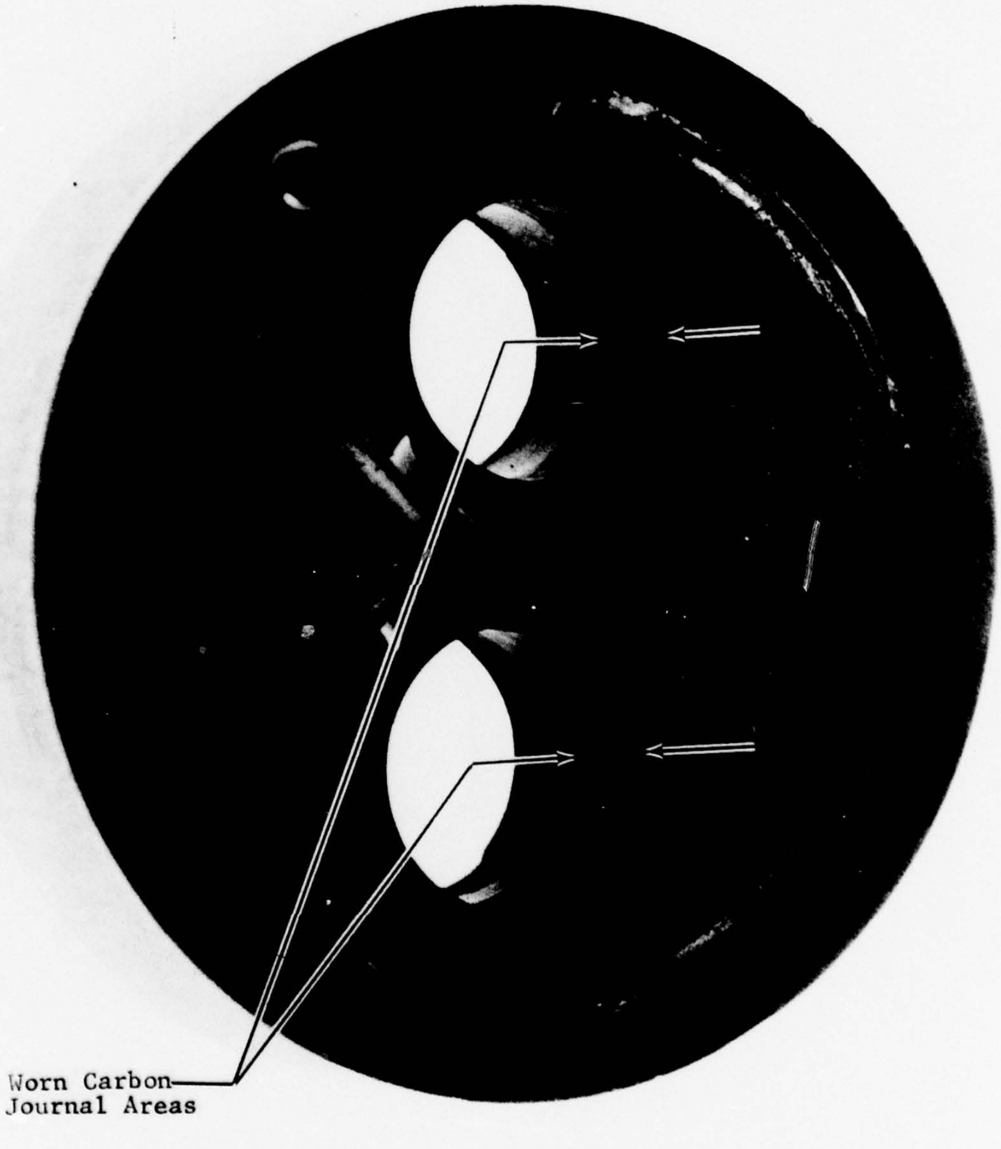


Figure 71. Scavenge Pump Front Bearing Assembly

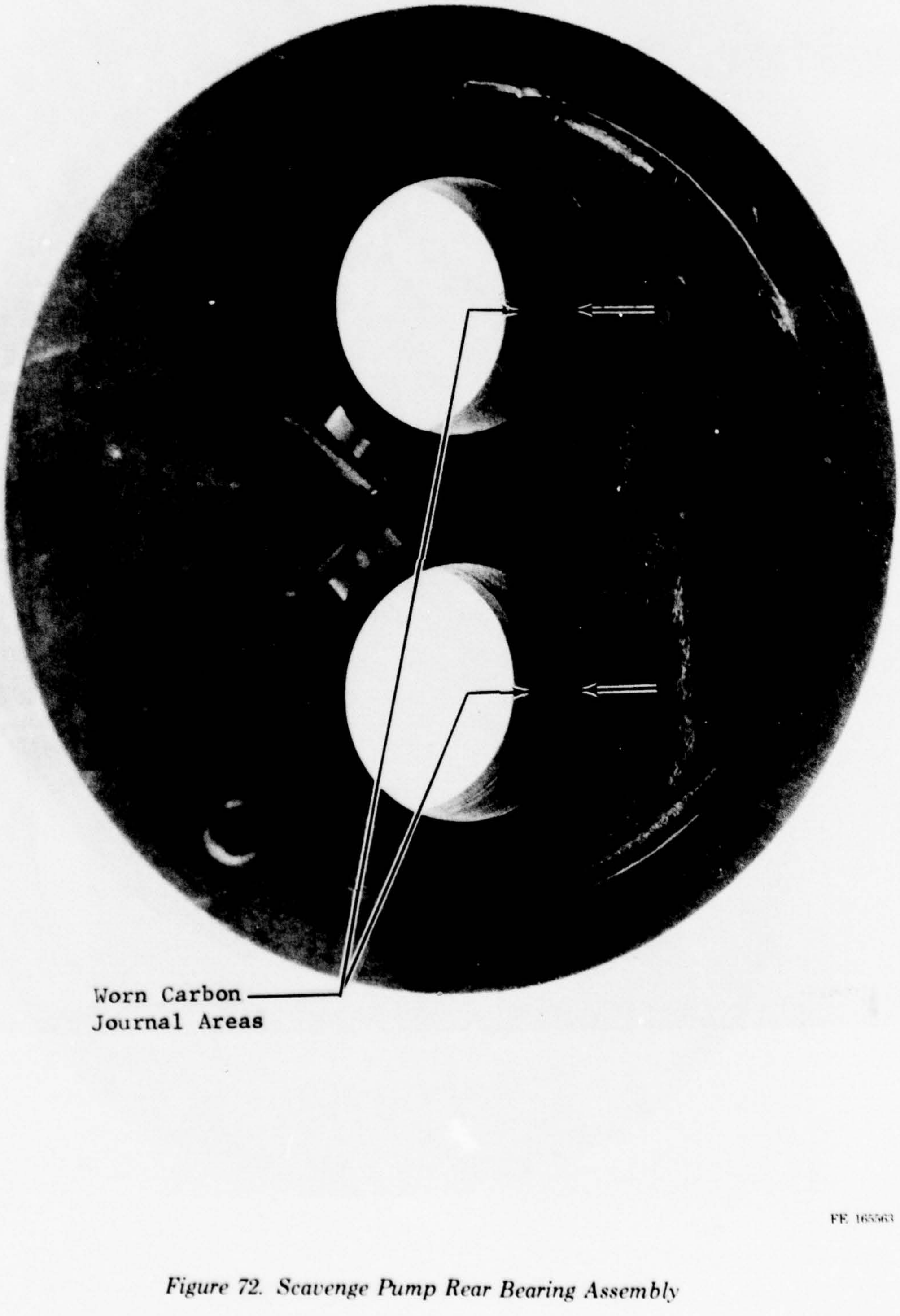
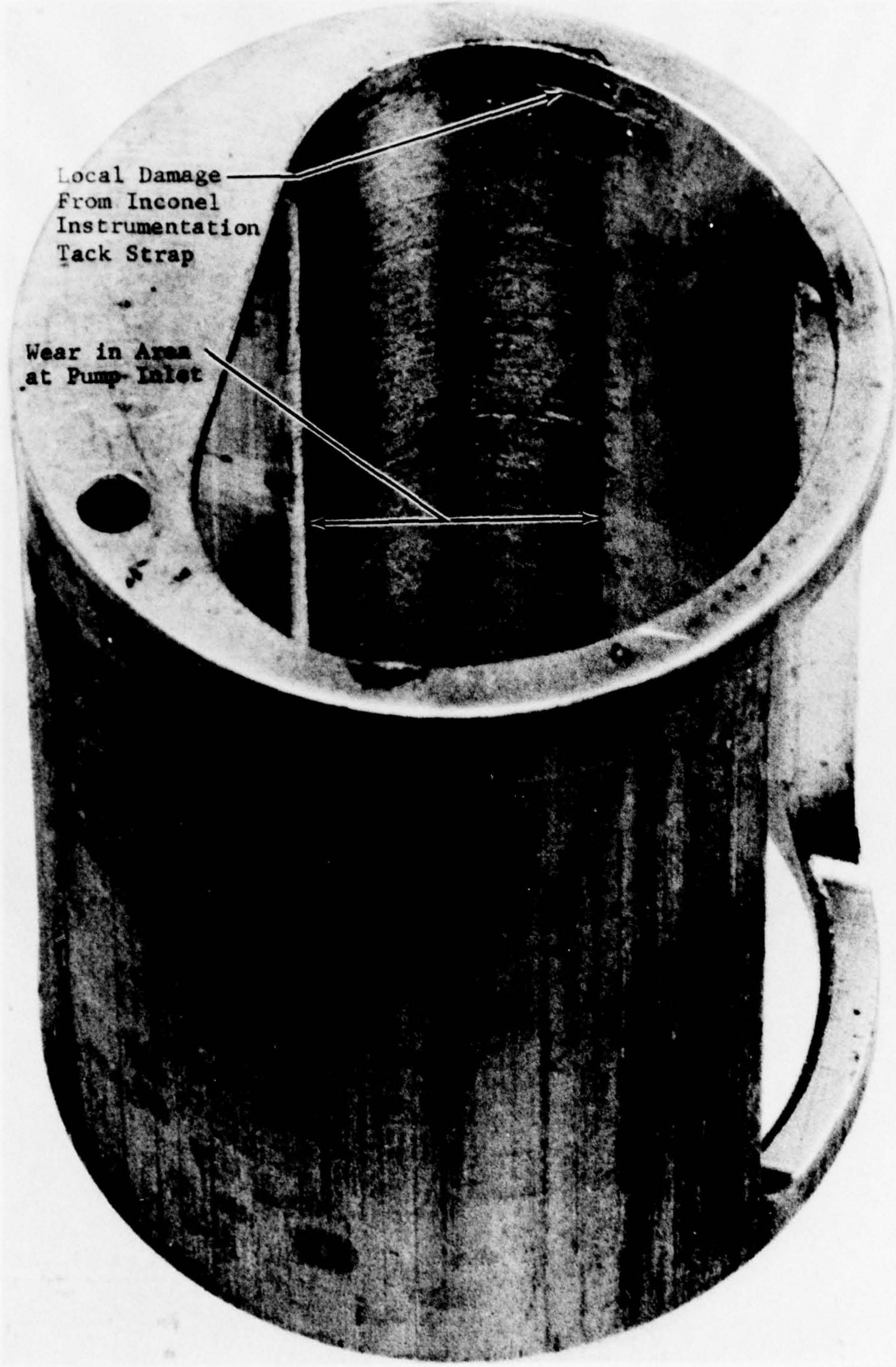
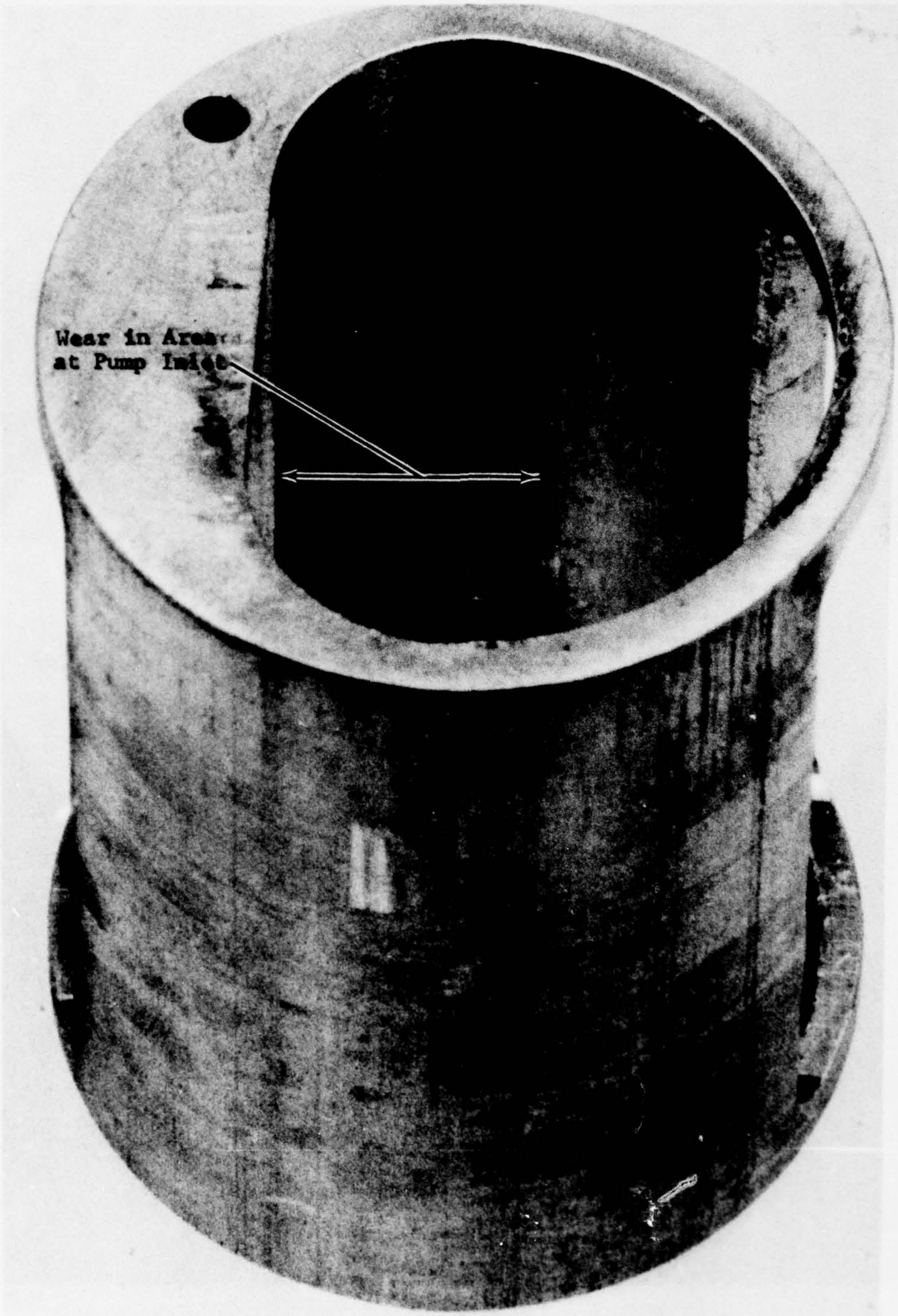


Figure 72. Scavenge Pump Rear Bearing Assembly



FE 165564

Figure 73. Supply Pump Sleeve, Package No. 1



FE 163565

Figure 74. Supply Pump Sleeve, Package No. 2

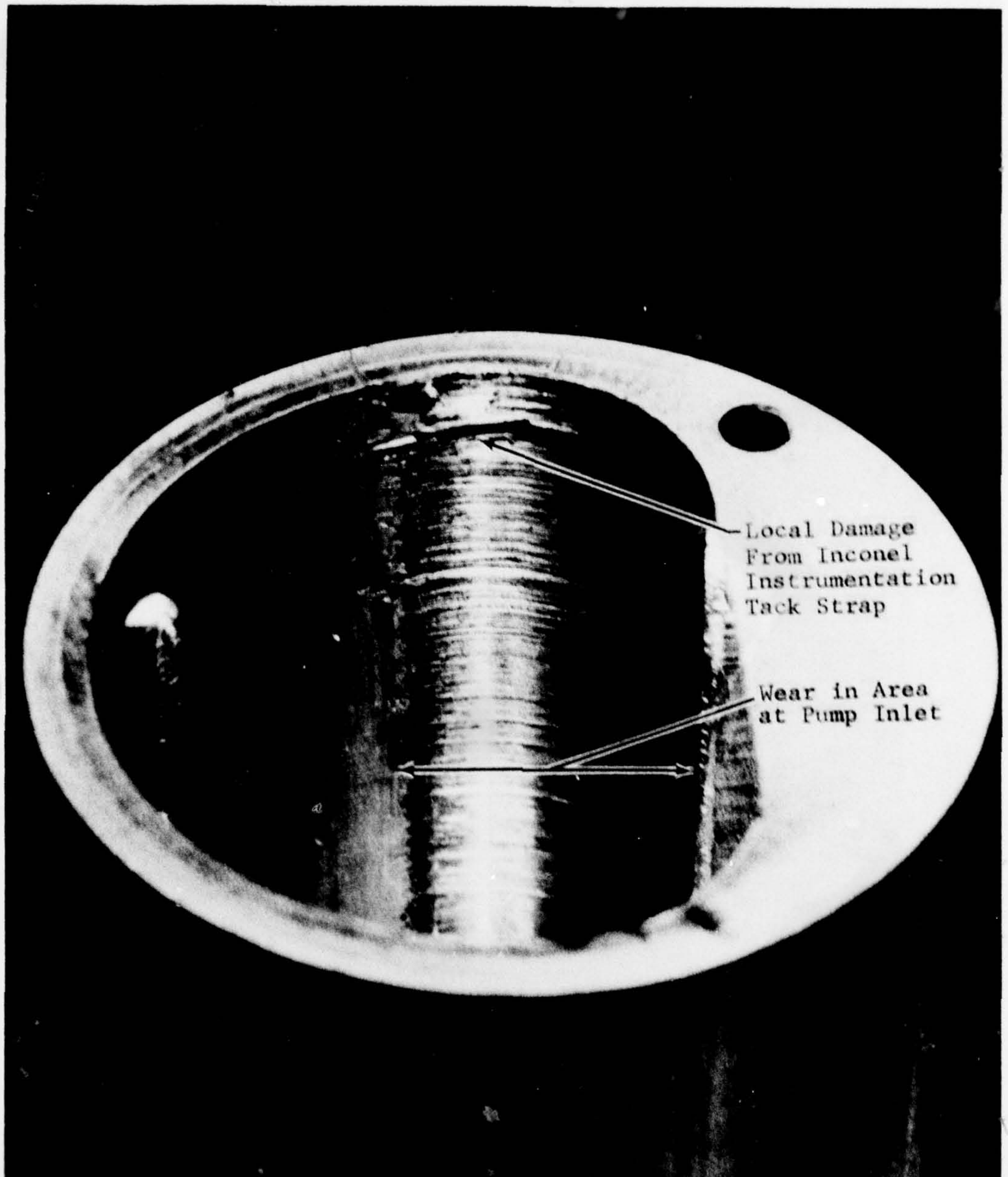


Figure 75. Scavenge Pump Sleeve

SECTION VI
CONCLUSIONS

The selected compartmental lubrication system concept provides for reduced vulnerability by locating major lubrication system components within an otherwise conventional bearing compartment. The 50-hour system endurance test substantiated the compartmental lubrication system concept. Consequently, this concept may be seriously considered for future, advanced gas turbine engines in which the lubrication system design criteria are weighted in favor of vulnerability, maintainability, and reliability considerations.

The system test program substantiated the technology considerations involved in the concept design by demonstrating the following:

- High-speed oil pump (both supply and scavenge) performance verification at two and one-half times conventional engine pump speeds.
- Feasibility of high-speed drive gear train in a compact bearing compartment.
- Capability to successfully deaerate labyrinth seal air leakages in excess of three times that of conventional engines within a small volume oil tank.
- Capability to properly scavenge a modified bearing compartment (in which a high-speed oil pump, drive train and oil tank, are installed) without any increase in lubrication system heat generation or oil foaming due to mechanical churning of the oil.

A comparative analysis with the baseline F100-PW-100 engine indicated that significant improvements are possible in vulnerability, maintainability, reliability, and frontal areas. The following results were obtained:

- Vulnerability — reduced 28.8 percent
- Maintainability — reduced 5756 maintenance man-hours per million engine flight hours
- Reliability — 962 fewer part discrepancies per million engine flight hours
- Frontal area — reduced 80 square inches.

The analyses and trade studies conducted indicate that labyrinth mainshaft seals, when used with properly sized scavenge pumps in conjunction with capped bearing compartments to limit air leakages, provide a feasible compartmental sealing configuration in advanced, high-speed engine applications where rotor speeds preclude the use of face seals. The system tests verified the feasibility of deaerating the air leakages associated with this configuration. Application of lift-off type mainshaft seals in a high-speed environment is an unproven approach for tomorrow's engine design whereas the labyrinth seal/scavenge pump system is a technically solid candidate for consideration in future high-speed mainshaft sealing applications.

SECTION VII

RECOMMENDATIONS

In future advanced engine design applications, such as RPV and VSTOL, lubrication system components will have reduced available space. Oil supply and scavenge pumps, associated drive gear trains and oil tank volume will, by necessity, have to be smaller than current conventional components. This will require higher speed pumps and gear trains and improvements in oil deaeration and compartmental scavenging. The full-scale rig tests conducted in the final phase of this program successfully demonstrated a compartmental lubrication system concept which meets those requirements. In future engine design efforts in which the criteria of vulnerability, maintainability, reliability, and frontal area are heavily weighted, it is recommended that lubrication system trade studies be conducted on a compartmental concept basis to determine the best system to meet design objectives. These studies should be performed early in the engine design phase while the basic engine configuration is still flexible to accept the results of the lubrication system studies.

It is further recommended that additional compartmental lubrication system studies be conducted in which the criteria of survivability is heavily weighted for system quantitative analyses. These studies should include analyses involving oil-mist lubrication systems as supplemental systems to conventional pump fed configurations.

APPENDIX A
COMPONENT SIZING SUMMARY

1. GENERAL

Lubrication system components sizes for each of the evaluated schemes are presented in this appendix.

2. SCHEME I

a. Oil Tank

The oil tank size is limited by the constraints of the No. 2-3 compartment boundaries. A design goal of a 3-gal capacity was initially set. Current mechanical design studies have indicated that the oil tank size for this scheme is 1.82 gal. Additional comments regarding tank capacity are discussed in design considerations.

b. Oil Supply Pump

The oil supply pump size was scaled from a 10,000-rpm ST9 gear pump. Scaling the element size to meet a 150 lb/min (250°F) oil flow requirement resulted in a 2.996-in. gear width. All pumping elements in the lubrication pump use 9-tooth/16-pitch gears, approximately ¾ in. in diameter.

c. Oil Scavenge Pumps

The scavenge elements run at 10,000 rpm and are scaled from the ST9 gear pump (discussed above). The No. 2-3 and 4 scavenge pumps were sized to twice the volumetric oil flowrate of their respective bearing compartments. This criterion was applied to compartments that are breathed. The resulting widths for the No. 2-3 and 4 scavenge elements were 3.54 and 1.558 in. respectively.

The No. 1 and 5 scavenge elements were sized to prevent compartmental oil loss during transient operation on deceleration. This sizing criterion required the No. 1 scavenge element to have six times the volumetric flow capacity of the compartmental oil flow. The No. 5 scavenge element was sized 12 times the compartmental oil flow capacity. The resulting element widths were 1.582 and 2.804 in. respectively for the No. 1 and 5 scavenge pumps.

d. Can Deaerator

This component remained the same size as its F100-PW-100 baseline counterpart which is approximately 7.7 in. long with a 3-in. diameter.

e. Oil Filter

The oil filter element volume remained the same as for the F100-PW-100 baseline system, (11.6 in.^{3})

f. Breather Pipes

The No. 4 and oil tank breather lines were 1-in. diameter.

g. Deoiler

The gearbox-mounted deoiler size remained the same as the baseline F100-PW-100, 5.7 in. in diameter.

h. Alternator

The alternator size was scaled upward from the F100-PW-100 baseline to reflect the lower operating speed resulting from the low rotor mount. The resulting size was 7 in. long by 5.7 in. in diameter.

3. SCHEME II

The oil supply and No. 2-3 scavenge pumps, alternator, and oil filter were the same as for Scheme I. The can deaerator and deoiler were the same size as that for the F100-PW-100 baseline.

a. Oil Tank

An attempt was made to design as large an oil tank capacity as possible into the No. 2-3 bearing compartment. Mechanical design studies showed that the oil tank size for this scheme was 2.5 gal.

b. Blowdown Plumbing

The No. 1, 4, and 5 compartment blowdown pipes were all 1.0 in. in diameter (OD).

c. Fuel/Oil Coolers

(1) Gas Generator Fuel/Oil Cooler

The gas generator fuel/oil cooler was a stainless steel-plate-fin heat exchanger in a single-pass, cross-flow configuration. The core dimensions, which do not include manifolds were:

Circumferential Wrap Length	=	20 in.
Length	=	11.9 in.
Thickness	=	0.8704 in.

These dimensions did not include manifolds.

(2) Augmentor Fuel/Oil Cooler

The augmentor fuel/oil cooler was also a stainless steel plate-fin heat exchanger in a single-pass, cross-flow arrangement. The core dimensions, which did not include manifolds, are as follows:

Circumferential Wrap Length	=	10 in.
Length	=	6.96 in.
Thickness	=	0.8704 in.

d. Air/Oil Cooler

The air/oil cooler was a finned-wall configuration which replaced the inner duct fairing. Its dimensions were as follows:

Circumferential Wrap Length	=	50 in.
Length	=	14 in.
Finned Surface	=	1 by 1/8 in.
Spacing Between Fins	=	1/8 in.
Fin Length (in direction of flow), Staggered	=	2 in.
Total Number of Fins	=	1372

4. SCHEME III

a. Oil Tank

No oil tank was required; each bearing compartment was an oil sump.

b. Oil Supply Pumps

Pump sizes are based on a vane pump design speed of 5000 rpm and a vane element diameter of 1.25 in. Journal bearing radius was 0.268 in.; journal length was 0.500 in. Housing thickness was 0.125 in. The vane width for each supply pump was:

Compartment Number	*Vane, Width, in.*
1	0.229
2-3	1.534
4	0.675
5	0.203

c. Oil Scavenge Pumps

Not required.

d. Alternator

Same as for Scheme I.

e. Oil Filter

Filter element volumes were:

Compartment Number	*Element Volume, in.^{3}*
1	1.00
2-3	6.75
4	2.97
5	0.89

f. Breather Pipes

All compartment breather pipes were 0.750 in. Manifold pipes combining all compartment air leakages were 1.00 in.

g. Deoiler

The required deoiler size was a function of speed. Figure A-1 illustrates this characteristic bivariately with air pressure drop across the deoiler. The selected sizes are shown superimposed on this figure and summarized below:

Compartment Number	*Deoiler Diameter, in.*
1	4.46
2-3	5.6 (Same as F100-PW-100 Baseline)
4	3.8
5	4.46

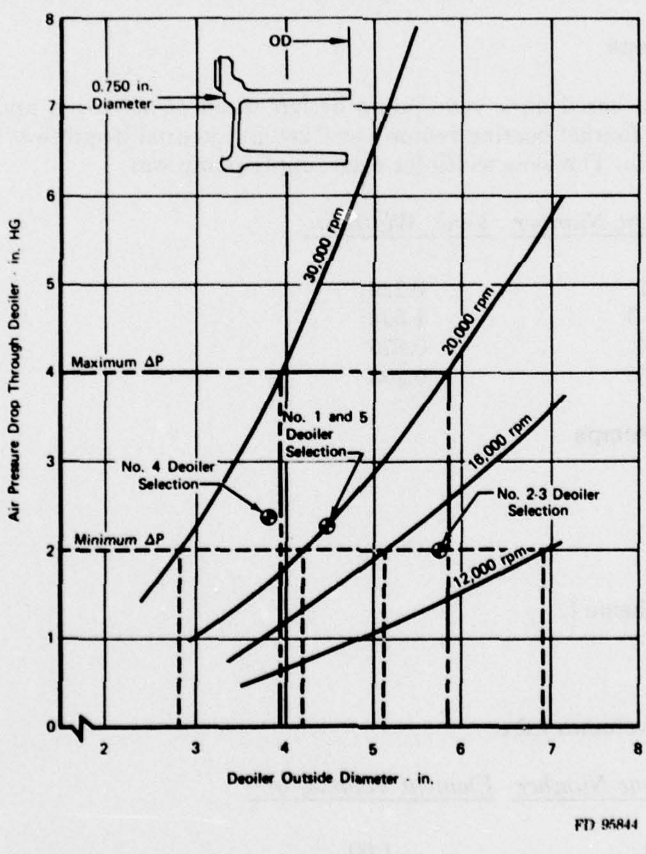


Figure A-1. Deoiler Size vs Deoiler Speed

h. Heat Pipes

The heat pipe sizes and arrangement are shown in Figure 5. The evaporators (located within the compartment) were shell and tube configurations with tubes, arranged as shown, resulting with a shell diameter of 4.2 in. Tube diameters were 0.100 in., with a 0.010-in. wall thickness. The tube wicks were 0.006 in. thick.

The ram air condenser was a series of shell and tube coolers with the air flowing through the tubes. A total of 600 tubes were used in the ram air condenser.

The augmentor fuel condenser was a series of shell and tube coolers with the fuel flowing through the tubes. A tube through the center core allowed part of the fuel to bypass the condenser during high fuel flow maximum augmentation conditions. The augmentor fuel condenser used 200 tubes that had a 0.100-in. diameter and 0.010-in. wall thickness.

The gas generator fuel condenser was a series of shell and tube coolers with the fuel flowing through 600 tubes.

An adiabatic intermediate media transfer tube connected the evaporators with the condenser for each compartment. These tubes provided a flowpath for the steam to travel from the evaporators to the condensers. Wicks inside the tubes provided the path for the water to be transferred from the condensers back to the evaporators. Transfer tube sizes were as follows:

Compartment Number	*Tube OD, in.*	*Wick Thickness, in.*
1	0.250	0.021
2-3	0.793	0.082
4	0.673	0.062
5	0.350	0.034

All condensers and evaporators were of stainless steel construction.

5. SCHEME IV

The can deaerator, oil filter, deoiler, air/oil, and fuel/oil coolers were the same as those in the F100-PW-100 baseline. The alternator was the same as that for Scheme I.

a. Oil Tank

The removal of the towershaft from the No. 2-3 compartment location provided maximum oil tank capacity in this lubrication scheme. Mechanical design studies showed the oil tank capacity for this scheme to be 3.03 gal.

b. Oil Supply Pumps

The main oil pump was the same as that in the baseline F100-PW-100. No boost oil pump was required.

c. Oil Scavenge Pumps

The scavenge pumps were 7-tooth/6-pitch gear elements. The element widths were as follows:

Compartment Number	*Width, in.*
1	0.578
2-3	3.08
4	0.895
5	1.09

6. SCHEME V

a. Oil Tank

The design goal was to package as large a tank capacity in the No. 2-3 bearing compartment as possible. Mechanical design studies showed the oil tank capacity for this scheme to be 1.82 gal.

b. Oil Pump

The oil pump was a positive displacement vane type with two stacks of elements. The supply stack ran at 5000 rpm and consisted of the main, boost, and No. 5 scavenge elements. The other stack included the No. 1, 2-3, and 4 scavenge elements and ran at 3200 rpm. Reduction gears provided the drive ratio off the towershaft/gear train. Pertinent envelope dimensions are shown in Figure A-2.

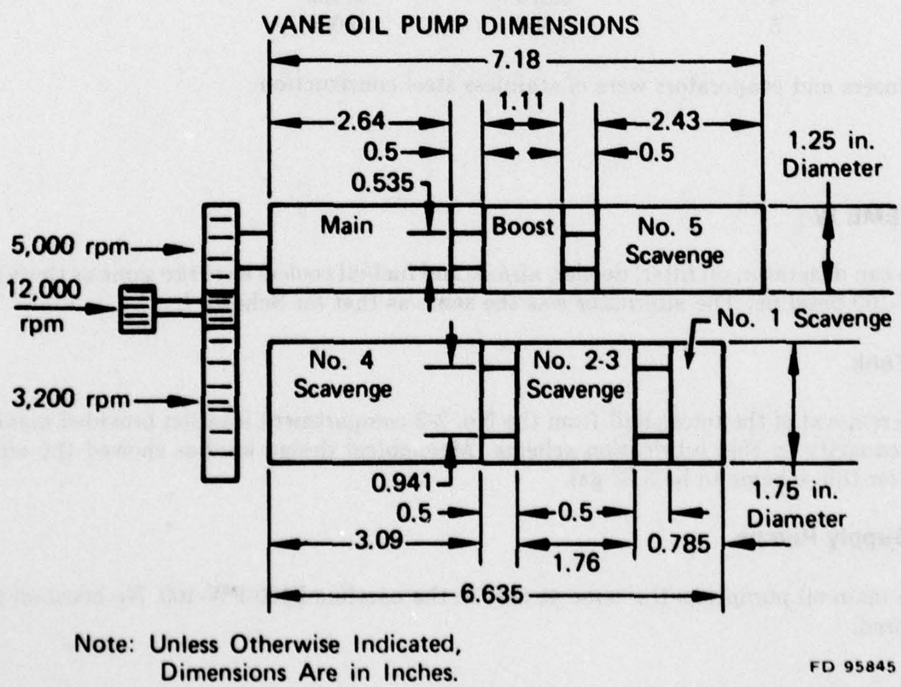


Figure A-2. Vane Oil Pump Dimensions

c. Centrifugal Oil Filter/Deoiler

The centrifugal oil filter/deoiler was designed to serve a dual function. This device filters the oil by centrifuging the contaminants out radially and, in addition, separates the air from the oil by providing vent holes and passages in the rotating shaft for the air to pass through on the way to the breather valve. A detailed discussion of this device is presented in the design considerations, including a summary of performance and geometry.

The gas generator and augmentor fuel/oil coolers and the air/oil coolers were the same as those in Scheme II. The alternator was the same as Scheme I. Plumbing and chip detectors were the same as those in the baseline F100-PW-100.

APPENDIX B
VULNERABLE AREA CALCULATIONS

Table B-1 shows a summary of the Δ vulnerable areas compared to the baseline engine for each of the six views for "A" and "B" kills using 30- and 50-caliber projectiles striking lubrication system components at 1500 and 2500 ft/sec. Table B-2 shows the "A" and "B" kill values averaged together and presented as a percentage of the baseline (F100-PW-100) vulnerable areas for each of the six views. These values were then multiplied by the probability of a hit from each direction (view factor) and then summed up for "A" and "B" kills at the bottom of Table B-2 for each scheme. The "A" and "B" kill values were then averaged, and the ratio of this value for the best scheme over a given scheme provided the comparison to best scheme factor for that scheme.

TABLE B-1.
VULNERABLE AREA (Δ OF BASELINE, IN.^{2}) OF COMPARTMENTAL LUBRICATION SYSTEM

		A	*A*	*A*	*A*	← *Kill* →	*B*	*B*	*B*	*B*
		30	*50*	*30*	*50*	← *Cal* →	*30*	*50*	*30*	*50*
View	*Scheme*	*1500*	*1500*	*2500*	*2500*	← *ft/sec* →	*1500*	*1500*	*2500*	*2500*
Front	I	+2.8	−2.1	+3.9	+4.7		−154.2	−154.2	−145.6	−143.2
	II	−5.6	−7.1	−5.6	−2.0		−180.8	−177.4	−173.4	−168.2
	III	+1.3	−7.4	−5.2	−5.5		−144.0	−144.0	−133.5	−133.8
	IV	+32.3	+27.4	+35.5	+34.1		−8.0	−8.0	+1.3	+1.3
	V	−5.6	−7.1	−5.6	−2.0		−147.4	−144.0	−144.0	−134.8
Rear	I	+8.1	+8.1	+8.1	+8.1		−97.1	−97.1	−101.2	−102.9
	II	+1.4	+11.3	+4.4	+16.5		−65.0	−51.5	−64.7	−49.7
	III	−4.3	+0.7	−4.4	+1.8		−137.3	−137.3	−141.4	−143.1
	IV	+11.5	+14.3	+18.6	+26.2		−58.0	−58.0	−55.7	−53.0
	V	+1.4	+11.3	+4.4	+16.5		−40.3	−26.8	−40.0	−25.0
Top	I	+53.8	+55.8	+49.7	+53.6		+54.1	+38.6	+50.5	+23.1
	II	+181.6	+183.6	+177.5	+181.4		+135.5	+120.8	+134.0	+109.9
	III	+52.8	+56.7	+48.1	+51.6		−273.1	−265.6	−262.6	−257.6
	IV	+69.0	+88.4	+72.0	+90.6		+244.2	+249.3	+261.1	+260.4
	V	+181.6	+183.6	+177.5	+181.4		+75.9	+139.8	+148.2	+120.7
Bottom	I	−87.8	−92.4	−80.5	−83.9		−296.4	−264.0	−290.6	−309.0
	II	−88.7	−90.8	−84.5	−54.3		−359.0	−323.3	−357.9	−323.0
	III	−88.7	−92.8	−85.7	−82.6		−379.2	−323.3	−372.2	−358.4
	IV	−90.6	−95.3	−85.8	−90.8		−303.8	−246.9	−283.5	−247.2
	V	−88.7	−90.8	−84.5	−50.8		−304.9	−277.1	−308.8	−285.8
Left Side	I	−7.5	−18.2	−10.2	−22.7		−51.0	−44.0	−49.0	−74.2
	II	−13.5	−24.0	−16.0	−28.5		+27.7	+18.9	+13.2	−36.4
	III	+27.1	+25.7	+23.2	+21.2		−292.1	−278.6	−288.0	−303.7
	IV	−0.1	−10.0	−4.1	−15.3		+97.9	+117.5	+93.1	+77.1
	V	−16.4	−29.1	−17.1	−30.3		+120.8	+127.9	+121.1	+95.8
Right Side	I	−47.8	−45.9	−49.9	−49.4		−382.6	−375.1	−377.0	−348.5
	II	−54.1	−52.2	−56.2	−55.7		−306.3	−306.9	−308.4	−307.4
	III	−2.4	−2.5	−5.7	−6.0		−623.3	−599.8	−605.9	−568.5
	IV	−40.4	−37.7	−43.8	−42.0		−282.6	−253.0	−264.6	−221.3
	V	−56.9	−57.4	−55.6	−54.7		−246.0	−226.1	−231.0	−199.3

TABLE B-2.
VULNERABLE AREA — AVERAGE OF PERCENT OF BASELINE COMPARTMENTAL LUBRICATION SYSTEM

View	*Scheme*	*View Factor, %*	*A Kill Average*	*A Kill Average Times Factor*	*B Kill Average*	*B Kill Average Times Factor*	*A and B Average With Factor*
Front	I	5	108.0	5.4	41.7	2.1	
	II		83.5	4.2	31.7	1.6	
	III		89.7	4.5	46.0	2.3	
	IV		202.7	10.1	98.5	4.9	
	V		83.5	4.2	44.7	2.2	
Rear	I	15	142.2	21.3	58.2	8.7	
	II		137.0	20.5	75.5	11.3	
	III		89.0	13.3	41.5	6.2	
	IV		186.7	28.0	76.2	11.4	
	V		137.0	20.5	86.2	12.9	
Top	I	10	151.2	15.1	108.0	10.8	
	II		275.2	27.5	124.0	12.4	
	III		150.7	15.1	48.5	4.8	
	IV		175.5	17.5	149.0	14.9	
	V		275.2	27.5	123.2	12.3	
Bottom	I	30	28.2	8.5	57.2	17.2	
	II		31.7	9.5	49.2	14.8	
	III		26.2	7.9	47.0	14.1	
	IV		24.2	7.3	59.5	17.8	
	V		32.5	9.7	56.5	16.9	
Left Side	I	20	90.2	18.0	90.2	18.0	
	II		86.0	17.2	101.2	20.2	
	III		118.2	23.6	46.5	9.3	
	IV		95.7	19.1	118.0	23.6	
	V		83.7	16.7	121.7	24.3	
Right Side	I	20	72.2	14.4	56.0	11.2	
	II		69.0	13.8	64.2	12.8	
	III		97.7	19.5	29.7	5.9	
	IV		76.5	15.3	70.0	14.0	
	V		67.7	13.5	73.7	14.7	
Total	I	100		82.7		68.0	75.3
	II			92.7		73.1	82.9
	III			83.9		42.6	63.2
	IV			97.3		86.6	91.9
	V			92.1		83.3	87.7

APPENDIX C
MAINTAINABILITY AND RELIABILITY CALCULATIONS

Table C-1 shows a component breakdown for each scheme of the Δ maintenance man-hours (MMH), Δ part discrepancies per million engine flight hours (EFH), and the ΔMMH per million EFH compared to the baseline F100-PW-100 engine. Note that a negative value for Δ part discrepancies per million engine flight hours means that this scheme has better reliability than the baseline F100-PW-100. All five schemes required more total maintenance man-hours per million engine flight hours than the baseline. Consequently, they received positive values for ΔMMH per million EFH and lower maintainability ratings than the baseline engine.

TABLE C-1
COMPARTMENTAL LUBRICATION SYSTEM RELIABILITY AND MAINTAINABILITY CHART (COMPARISONS TO BASELINE)

Component	Scheme I 1	Scheme I 2	Scheme I 3	Scheme II 1	Scheme II 2	Scheme II 3	Scheme III 1	Scheme III 2	Scheme III 3	Scheme IV 1	Scheme IV 2	Scheme IV 3	Scheme V 1	Scheme V 2	Scheme V 3
Alternator	+0.5	+34	+427	0.1	34	262	+0.1	+34	+262	+0.1	+34	+262	+0.1	+34	+262
Main Gearbox	+1.0	−90	−281	+1.0	−90	−281	+1.0	−90	−281	+1.0	−38	+270	+1.0	−71	−281
Main Oil Pump	+31.0	+74	+7,004	+31.0	+99	+7,904	−5.0	−770	−3,850	0	0	0	+29.8	+99	+7,474
Boost Pump	−4.1	−126	−517	−4.1	−126	−517	Included in Main Oil Pump			0	0	0	+30.1	+73	+7,323
No. 1 Scavenge Pump	+2.6	+63	+769	−4.4	−126	−554	Included in Brg Compt			0	0	0	+35.8	+63	+7,045
No. 2-3 Scavenge Pump	+30.9	+88	+7,061	+30.9	+88	+7,061	Included in Brg Compt			0	0	0	+35.1	+88	+7,437
No. 4 Scavenge Pump	+31.9	+63	+6,287	−4.1	−126	−517	Included in Brg Compt			0	0	0	+36.1	+63	+7,082
No. 5 Scavenge Pump	+9.8	+73	+2,272	−4.1	−126	−554	Included in Brg Compt			0	0	0	+35.8	+63	+7,045
Oil Filter	+2.6	−27	+1,373	+2.6	0	+1,508	−2.4	−580	−1,392	0	0	0	0	0	0
Oil Tank	+33.7	−200	+888	−2.3	−200	−522	−2.4	−240	−576	−2.3	−200	−552	+15.7	−200	+168
Deaerator	+14.7	0	+44	+14.4	0	+44	Included in Brg Compt			+14.4	0	+44	Included in Brg Compt		
Fuel/Oil Coolers	+0.5	0	+95	+6.9	0	+1,311	0	0	0	0	0	0	+6.9	0	+1,311
Air/Oil Coolers	0	0	0	+20.7	0	+8,487	0	0	0	0	0	0	+20.7	0	+8,487
Oil Filter Bypass Valve	+2.6	+27	+70	Included in Oil Filter			Included in Oil Filter			Included in Oil Filter			Included in Oil Filter		
Oil Pressure Bypass Valve	0	+25	0	Included in Main Oil Pump			Included in Main Oil Pump			Included in Main Oil Pump			Included in Main Oil Pump		
No. 1 Bearing Compartment	0	+40	+424	0	−240	−2,544	+3.3	+565	+9,932	0	0	0	0	0	0
No. 2-3 Bearing Compartment	0	0	0	0	+35	+1,120	+1.0	+521	+20,299	0	−403	−12,896	0	+40	+1,280
No. 4 Bearing Compartment	0	0	0	0	−378	−13,608	+4.0	+565	+25,672	+26.0	+348	+41,544	0	0	0
No. 5 Bearing Compartment	0	0	0	0	−320	−5,664	+0.3	+565	+10,446	0	0	0	0	0	0
Towershaft	0	0	0	0	0	0	0	0	0	+2.2	+123	+1,884	0	0	0
Inlet Fan Module	−2.3	0	−431	−2.3	0	−431	−2.3	0	−431	−2.3	0	−431	−2.3	0	−431
Intermediate Case	0	0	0	0	0	0	0	0	0	−14.3	0	−2,011	0	0	0
High Compressor R&S Assembly	0	0	0	0	0	0	0	0	0	+14.1	0	+48,715	0	0	0
Fan Ducts	0	0	0	0	0	0	0	0	0	+14.1	0	+1,449	0	0	0
Plumbing	0	0	0	0	0	0	+2.5	+100	+250	0	0	0	0	0	0
Diffuser Case	0	0	0	0	0	0	0	0	0	+17.1	+50	+15,975	0	0	0
Total		+44	+25,485		−1,476	+2,475		+670	+60,773		−86	+94,253		+252	+54,202

1 ΔMaintenance Man-Hours (MMH)
2 ΔPart Discrepancies per Million EFH
3 ΔMMH per Million EFH

AD-A060 172 PRATT AND WHITNEY AIRCRAFT GROUP WEST PALM BEACH FL G--ETC F/G 11/8
COMPARTMENTAL LUBRICATION SYSTEM.(U)
JUN 78 E M BEVERLY F33615-75-C-2075
UNCLASSIFIED PWA-FR-9555 AFAPL-TR-78-32 NL

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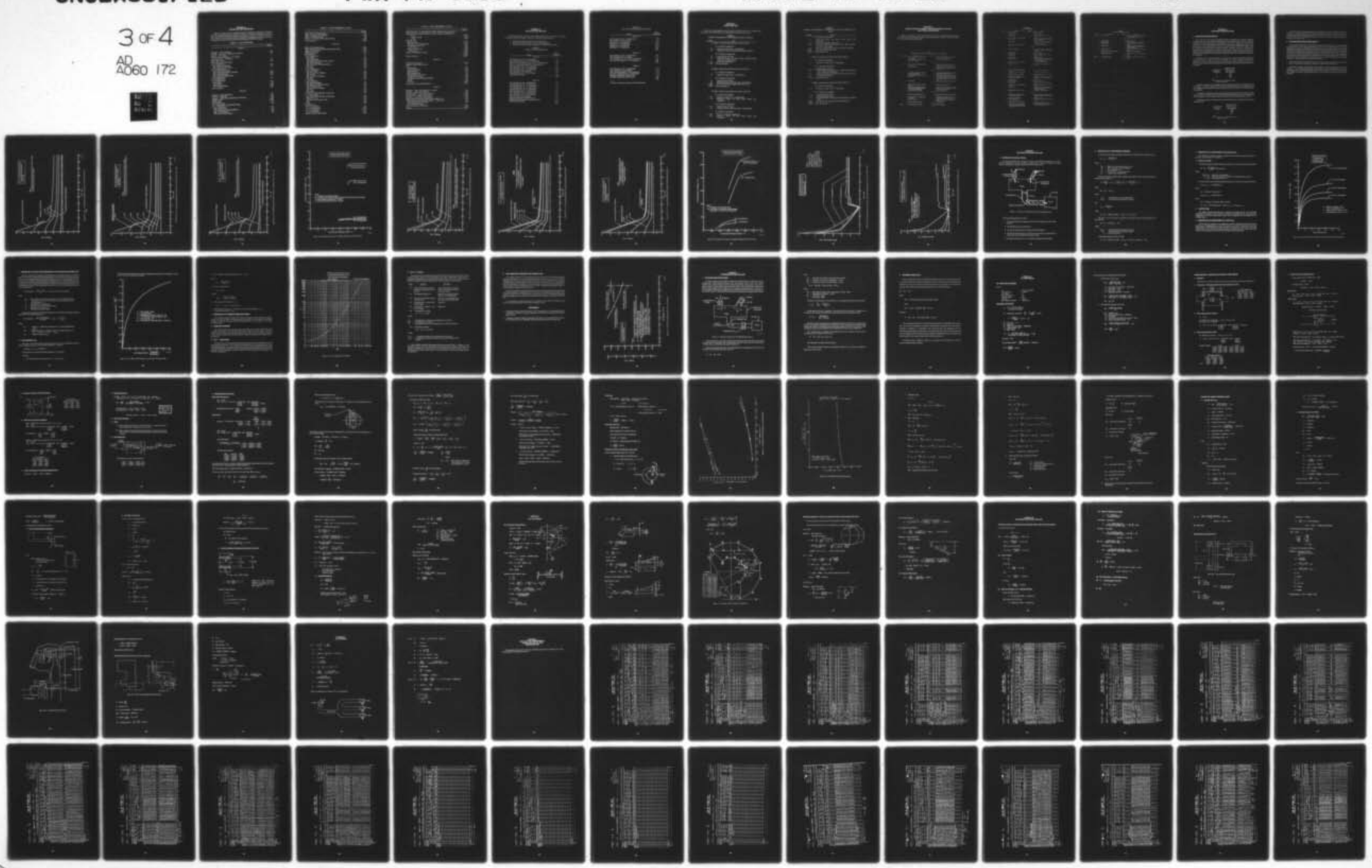


APPENDIX D
ACQUISITION COST BREAKDOWN

Table D-1 presents the cost of lubrication system components compared to the baseline F100-PW-100 engine for Schemes I through V. A positive Δ means that the component costs more than the baseline F100-PW-100 component and a negative Δ reflects a reduction in the cost of that component. Note in Scheme II that reverting back to the baseline cooler system results in a less expensive lubrication system than that of the baseline F100-PW-100 engine.

TABLE D-1. COST SUMMARIES

	ΔDollars
Scheme I	
Alternator — Factor for Size Δ	+314
Oil Tank — Factor for Configuration and Size Δ	−369
Gearbox — No. Change	—
Strainers and Chip Detectors — No Change	—
Coolers and Filter — No Change	—
Delete: Boost Pump	−317
Delete: Main Oil Pump Housing	−780
No. 1 Compartment —	
Add: 2 Gears and Housing	+250
Add: Alternator Can Housing	+50
No. 2-3 Compartment —	
Add: 6 Drive Gears at $65	
Add: 4 Bearings at $50 and 1 Housing	+350
Add: 2 Pump Housings	+350
Add: 2 Pump Housing Supports	+200
No. 4 Compartment —	
Add: Breather Line	+155
No. 5 Compartment —	
Add: 2 Gears	+130
Add: Housing	+100
Add: Housing Support	+120
Total Δ Scheme 1	+943
Scheme II	
Alternator — Same as Scheme I	+314
Oil Tank — Delete 80%	−1,476
Gearbox — Delete Gears and Bearings for Oil Pump	−628
Fuel/Oil Coolers	+3,308
Air/Oil Cooler	+3,453
Delete Boost Pump	−317
No. 1 Compartment — Alternator Housing	+50
No. 2-3 Compartment —	
Add: 3 Drive Gears	+195
Add: 2 Bearings and Bearing Housing	+175
Add: 1 Pump Housing	+175
Add: Pump Housing Support	+100

TABLE D-1. COST SUMMARIES (Continued)

	ΔDollars
Delete: Main Oil Pump Housing	−780
Delete: 3 Scavenge Pump Modules	−1,050
Add 4 1-in. Blowdown Lines	+450
Replace 4 Carbon Seals With Labyrinth Seals	−500
Total Scheme II Δ	+3,469
Scheme III	
Delete: 16 Lubrication Lines	−2,400
Delete: Main Oil Pump Housing and Scavenge Pumps	−1,830
Delete: Oil Tank (80%)	−1,476
Delete: Boost Pump	−317
Add Alternator	+314
No. 1 Compartment —	
Add: Deoiler	+219
Add: Deoiler Shaft	+150
Add: Deoiler Bearings, Housing, and Gear	+320
Add: Main Drive Gear	+135
Add: Oil Pump With Bypass and Gear	+350
Add: Filter	+276
Add: Sump	+150
No. 2-3 Compartment —	
Add: Drive Shaft	+125
Add: Housing and 2 Bearings	+220
Add: 4 Gears	+275
Add: Oil Pump with Bypass	+400
Add: Filter	+575
Add: Sump	+150
No. 4 Compartment —	
Add: Deoiler and Shaft	+369
Add: Bearings, Housing, and Gear	+320
Add: Main Drive Gear	+175
Add: Oil Pump with Bypass	+350
Add: Filter	+375
Add: Sump	+150
Pump Drive Shaft, Bearings, and Housing	+320
No. 5 Compartment —	
Add: Deoiler Shaft Bearings, and Housings	+689
Main Drive Gear	+225
Oil Pump with Bypass and Gear	+350
Filter	+276
Sump	+150
Evaporator Cost —	
No. 1 Compartment	+754
No. 2-3 Compartment	+4,879
No. 4 Compartment	+3,733
No. 5 Compartment	+1,094
4 Breather Lines at $300 Average	+1,200

TABLE D-1. COST SUMMARIES (Continued)

	ΔDollars
4 Heat Pipe Lines, 4 × average Oil Line ($150) +$250 for Each Union and 4 Charging Valves for Media at $250 ea, 3 service unions per compartment	
4 Lines	+2,400
12 Service Unions	+3,000
4 Valves	+1,000
Condenser Cost	
Ram Air Cooler	+12,593
Add $125 ea for 8 Special Unions	+1,000
Gas Generator Fuel Cooler	+3,496
Add 4 Unions	+500
Augmentor Fuel Cooler	+5,029
Add 8 Unions	+1,000
Subtract Bill-of-Material Coolers	−4,536
	−2,670
Scheme III Total Δ	+35,857
Scheme IV	
Oil Tank Like Scheme II	−1,470
Alternator Like Schemes I and II	+314
Add	
Splined Shaft	+225
Angle Case Adapter	+298
Heat Shielding	+254
Ball Bearing Sleeve	+95
Retaining Ring, Outer Case	+100
Compartment Housing	+500
Compartment Housing Bearing Support	+750
Diffuser	+225
Fan Case	+388
Total Increase From Bill-of-Material Δ	+1,679
Scheme V	
Alternator — Same as for Schemes I, II, and IV	+314
Oil Tank — Same as for Scheme I	−369
Fuel/Oil Coolers — Same as for Scheme II	+3,308
Air/Oil Cooler — Same as for Scheme II	+3,453
Oil Pump — Vane Type, With Support	+675
Gearbox Delete Deoiler and Filter, Add Combination +Δ	+262
Add Oil Pump Drive System Ref Scheme I — Add 5 Gears	+325
Add 6 Bearings and 2 Housings	+540
Pump Housing Tradeoff	—
Boost Vane Pump vs Gear Pump	+675
Total Δ	+9,183

APPENDIX E
LIFE CYCLE COST ANALYSIS

Table E-1 presents the Δ life cycle cost in millions of dollars for each scheme, compared to the baseline F100-PW-100 engine on the basis of the following ground rules:

1. Air superiority fighter application; 15-year life cycle
2. 1000 total engines including 15 percent uninstalled spares
3. 75 percent of installed engines operational; 25 flight hours per month.

TABLE E-1
LIFE CYCLE COST ANALYSIS

Change	*ΔLCC** *\$ (Millions)*
Scheme I	
Move Alternator to No. 1 Compartment	+1.2
Move Scavenge Pump to No. 1 Compartment	+0.4
Move 2-3 and 4 Scavenge Pumps to No. 2-3 Compartment	+1.9
Move No. 5 Scavenge Pump to No. 5 Compartment	+0.7
Move Oil Pump to No. 2-3 Compartment	+0.2
Move Oil Filter to No. 2-3 Compartment	+0.1
Move Oil Tank to No. 2-3 Compartment	−0.9
Move Gearbox to Top of Engine	−0.2
Move Fuel/Oil Cooler to Top of Engine	0
Eliminate Boost Pump	−0.5
	+2.9
Scheme II	
Move Alternator to No. 1 Compartment	+1.0
Scavenge Revisions, No. 1 Compartment	−1.2
Scavenge Revisions, No. 2-3 Compartment	+0.3
Scavenge Revisions, No. 4 Compartment	−2.1
Scavenge Revisions, No. 5 Compartment	−1.6
Move Oil Pump to No. 2-3 Compartment	+1.3
Move Oil Filter to No. 2-3 Compartment	+0.1
Move Oil Tank to No. 2-3 Compartment	−2.5
Move Gearbox to Top of Engine	−1.1
Redesign and Relocate Fuel/Oil Cooler	+5.0
Redesign Air/Oil Cooler and Locate Inside Duct	+5.3
Eliminate Boost Pump	−0.5
	+4.0

TABLE E-1
LIFE CYCLE COST ANALYSIS (Continued)

Change	ΔLCC* $ (Millions)
Scheme III	
Move Alternator to No. 1 Compartment	+ 0.9
Move Gearbox to Top of Engine	− 0.1
Redesign No. 1 Compartment	+ 9.8
Redesign No. 2-3 Compartment	+27.4
Redesign No. 4 Compartment	+16.9
Redesign No. 5 Compartment	+11.1
	+66.0
Scheme IV	
Move Alternator to No. 1 Compartment	+0.9
Mount Gearbox on Top of Engine	−0.1
Move PTO From No. 2-3 to No. 4 Compartment	+8.7
Move Oil Tank Inside No. 2-3 Compartment	−2.5
	+7.0
Scheme V	
Move Alternator to No. 1 Compartment	+ 1.0
Mount Gearbox on Top of Engine	+ 0.3
Move Oil Pump Inside No. 2-3 Compartment	+ 5.7
Move Oil Tank Inside No. 2-3 Compartment	− 1.0
Redesign and Relocate Fuel/Oil Cooler	+ 5.0
Redesign and Relocate Air/Oil Cooler	+ 5.4
	+16.4

*All Values Are Differentials Compared to the Baseline F100 Engine

APPENDIX F
WEIGHT ANALYSIS

Table F-1 shows the differential weight of all five candidate schemes on a component basis. Note that all five schemes had a total weight greater than the baseline engine.

TABLE F-1
WEIGHT COMPARISONS TO BASELINE F100-PW-100 ENGINE

ΔW, lb	Item
	L-231663-1 Compartmented Lubrication System Scheme I
	No. 1 Bearing Compartment
+ 8.6	Alternator Located in No. 1 Compartment
+ 7.9	Lubrication System, Scavenge Pump, Sump, and Filter
	No. 2-3 Bearing Compartment
− 3.3	Compartmental Oil Tank
− 1.3	Lubrication System, Oil Pump, Filter, Plumbing, Relief Valve, and Scavenge Pumps
+ 3.1	Revise Intermediate Case
+ 15.0	Total ΔW Scheme I
	L-231663-2 Compartmented Lubrication System Scheme II
	No. 1 Bearing Compartment
+ 7.6	Alternator Located in No. 1 Compartment
	No. 2-3 Bearing Compartment
− 2.1	Compartmental Oil Tank
− 3.1	Lubrication System, Oil Pump, Filter, and Plumbing
+ 58.6	Air Oil, Fuel Oil, and Augmentor Fuel Oil Coolers
+ 61.0	Total ΔW Scheme II
	L-231663-3 Compartmented Lubrication System Scheme III
	No. 1 Bearing Compartment
+ 9.2	Alternator Located in No. 1 Compartment
+ 6.5	Lubrication System, Oil Pump, Filter, Deoiler, and Evaporator
	No. 2-3 Bearing Compartment
− 13.0	Compartmental Oil Tank
− 8.5	Lubrication System, Oil Pump, Filter, and Evaporator
	No. 4 Bearing Compartment
+ 20.0	Revise No. 4 Bearing Compartment
+ 19.2	Lubrication System, Oil Pump, Filter, Deoiler, and Evaporator

TABLE F-1
WEIGHT COMPARISONS TO BASELINE F100-PW-100 ENGINE (Continued)

ΔW, lb	Item
	No. 5 Bearing Compartment
+ 4.1	*Lubrication System, Oil Pump, Filter, Deoiler, and Evaporator*
+ 3.0	*Revise No. 5 Bearing Compartment*
+ 53.5	*Increased Diameter for Forward Fan Duct, Rear Fan Duct, Combustor, and Diffuser*
+ 33.0	*Ram Air, Augmentor Fuel, and Gas Generator Condensers*
+ 23.0	*Heat Pipes and Breather Pipes*
+150.0	*Total W Scheme III*
	L-231663-4 Compartmented Lubrication System Scheme IV
	No. 1 Bearing Compartment
+ 7.6	**Alternator Located in No. 1 Compartment**
	No. 2-3 Bearing Compartment
− 7.2	**Compartmental Oil Tank**
+ 21.1	**Transfer Main Gearbox From No. 2-3 Bearing Compartment to No. 4 Bearing Compartment**
+ 53.5	**Increased Diameter for Forward Fan Duct, Rear Fan Duct, Combustor, and Diffuser**
+ 75.0	**Total ΔW Scheme IV**
	L-231663-5 Compartmented Lubrication System Scheme V
	No. 1 Bearing Compartment
+ 7.6	**Alternator Located in No. 1 Compartment**
	No. 2-3 Bearing Compartment
− 4.6	**Compartmental Oil Tank**
+ 3.1	**Revise Intermediate Case**
− 1.7	**Lubrication System, Oil Pump, Filter, Plumbing, and Relief Valve**
+ 58.6	**Air Oil, Fuel Oil, and Augmentor Fuel Oil Coolers**
+ 63.0	**Total ΔW Scheme V**

APPENDIX G
MANUFACTURING, ASSEMBLY, AND DEVELOPMENT ANALYSIS AND SYSTEM COMPROMISES

Table G-1 provides a list of manufacturing, assembly, and development difficulties associated with each of the five schemes, plus the baseline engine. Table G-2 provides a similar list for lubrication system compromises.

TABLE G-1
MANUFACTURING, ASSEMBLY, AND DEVELOPMENT CONSIDERATIONS

Scheme	*Item*	*Difficulty*	*Points*	*Remarks*
I	1	Difficult Assembly of Components Inside Compartments	−7	Compartments would probably have to be preassembled before installation in engine.
	2	Shaft Length Was Increased to Provide Drive for No. 5 Pump	−2	Decreases critical speed margin.
			$\Sigma = -9$	
II	1	Difficult Assembly of Components Inside Compartments	−5	Compartments would probably have to be preassembled before installation in engine.
	2	Internal Air-Oil Cooler	−7	Offset fin arrangement is difficult to fabricate. Internal cooler requires engine disassembly to remove cooler.
			$\Sigma = -12$	
III	1	Very Difficult To Assemble Components Inside the Compartments	−10	Compartments would probably have to be preassembled before installation in engine.
	2	Difficult Development Problem With Heat Pipe Concept	−10	Problem with continuous wick at mechanical joints between transfer pipes and condenser and evaporator. Also, a method must be developed to bond a 0.006 in. wick inside 0.100-in. tubes.
	3	Internal Coolers	−7	Requires engine disassembly to service coolers.
			$\Sigma =\ \ 27$	
IV	1	Towershaft Is Inclined and Longer Than Baseline	−3	Possible critical speed problem.
	2	Radial Load From Towershaft Is Removed from Main Shaft Ball Bearing. Ball and Roller Locations Interchanged.	−6	Possible rotor dynamics problem with radial location. Location of ball bearing will allow more axial play in compressor.
	3	All Pumps Stacked On One Shaft	−3	Require close tolerances.
			$\Sigma = -12$	
V	1	Combination Centrifugal Filter-Deoiler	−4	Must determine capability to filter and deaerate oil.
	2	Difficult Assembly of Components Inside Compartments	−9	Compartments would probably have to be preassembled before installation in engine.
	3	Internal Air-Oil Cooler	−7	Internal cooler requires engine disassembly to remove cooler.
	4	All Pumps Stacked On One Shaft	−1	Tolerance problems
			$\Sigma = -21$	
Baseline	1	All Pumps Stacked On One Shaft	−3	Tolerance problems
			$\Sigma = -3$	

TABLE G-2
SYSTEM COMPROMISES

Scheme	Item	Compromise	Points	Remarks
I	1	Undersized Oil Tank	−10	Inadequate deaeration.
	2	Internal Oil Pumps	−7	Difficult to service.
	3	Internal Oil Filter	−5	Difficult to inspect or replace.
	4	Internal Oil Tank	−5	No visual inspection of oil level. Difficult to service.
	5	No. 4 Compartment Air Vented Through Breather Pipe	−3	Pipe could coke and clog, causing oil spillage. Also breather pipe increases airflow into compartment. *Fire danger.*
	6	Internal Alternator	−5	Difficult to service. Size would have to be increased substantially for FAEC. Would not provide power during start.
	7	Airflow Opposes Oil Flow Down Tower-shaft	−3	Possible gravity scavenging problem.
			Σ = −38	
II	1	Blowdown Scavenge System	−7	These systems are prone to coking, high-air inflow, and oil spillage during transients. Fire hazard.
	2	Internal Oil Pumps	−5	Difficult to service.
	3	Internal Oil Filter	−5	Difficult to inspect or replace.
	4	Internal Oil Tank	−5	No visual inspection of oil level. Difficult to service.
	5	Undersize Oil Tank	−7	Inadequate deaeration.
	6	Internal Alternator	−5	Difficult to service. Size would have to be increased substantially for FAEC. Would not provide power during start.
	7	Airflow Opposes Oil Flow Down Tower-shaft	−3	Possible gravity scavenging problem.
	8	*Internal Air-Oil Cooler*	−4	Fan duct would have to be removed to service cooler.
			Σ = −41	
III	1	Each Compartment Must Be Serviced and Monitored	−10	Oil gage for each compartment, etc.
	2	Inadequate Oil Sump in No. 2-3 Compartment	−3	Approximately one-half required volume.
	3	Requires Ram Scoop for Air Condenser	−7	Engine-airframe interface, difficult to test cooler capacity.
	4	Mainstream Airflow Must Be Diverted to Enlarge No. 4 Compartment	−5	May require burner development. May cause bypass flow problems due to decreased fan duct area.
	5	Internal Oil Pumps	−10	Difficult to service.
	6	Internal Oil Filters	−8	Difficult to inspect or replace.
	7	Internal Oil Tank	−8	No visual inspection of oil level. Difficult to Service.
	8	Possible Freezing of Heat Pipe Media (Water)	−4	Will require antifreeze solution for −40°F operation.
	9	Internal Alternator	−5	Difficult to service. Size would have to be increased substantially for FAEC. Would not provide power during start.
	10	Airflow Opposes Oil Flow Down Tower-shaft	−3	Possible gravity scavenging problem.
			Σ = −33	
IV	1	Mainstream Airflow Must be Diverted to Enlarge No. 4 Compartment	−5	May require burner development and cause bypass flow problems due to decreased fan duct area.
	2	Towershaft In Hot Environment	−4	May result in coking.
	3	Scavenge Breather System	−4	Requires boost pump and oversized scavenge pumps.
	4	Airflow Opposes Oil Flow Down Tower-shaft	−3	Possible gravity scavenging problem.
	5	Oil Pump Mounted On Top of Engine Above Oil Tank Could Present Net Positive Suction Head (NPSH) Problems At Altitude	−3	Problem could be alleviated by increasing the breather pressurizing valve setting.

TABLE G-2
SYSTEM COMPROMISES (Continued)

Scheme	*Item*	*Compromise*	*Points*	*Remarks*
IV (Cont'd)	6	Internal Alternator	−5	Difficult to service. Size would have to be increased substantially for FAEC. Would not provide power during start.
			$\Sigma = -24$	
V	1	Undersized Oil Tank	−7	Inadequate deaeration.
	2	Internal Oil Pumps	−10	Difficult to service. Difficult plumbing problem.
	3	Internal Oil Tank	−5	No visual inspection of oil level. Difficult to service.
	4	Internal Alternator	−5	Difficult to service. Size would have to be increased substantially for FAEC. Would not provide power during start.
	5	Internal Air-Oil Cooler	−4	Fan duct would have to be removed to service cooler.
	6	Airflow Opposes Oil Flow Down Tower-shaft	−3	Possible gravity scavenging problem.
	7	Scavenge Breather System	−4	Requires boost pump and oversized scavenge pumps.
			$\Sigma = -38$	
Baseline	1	Scavenge Breather System	−4	Requires boost pump and oversized scavenge pumps.
			$\Sigma = -4$	

APPENDIX H
SCAVENGE SYSTEM ANALYSES

1. BLOWDOWN SYSTEM RESULTS

An evaluation of the scavenge blowdown system was performed to determine required blowdown line sizes and resulting compartmental seal leakages. The analyses conducted considered both carbon face type seals and labyrinth seals in the numbers 1, 4, and 5 bearing compartments. The compartmental environmental pressures and temperatures during engine decelerations were taken from FX205-21 which is a typical baseline (F100-PW-100) engine. Using the blowdown scavenge transient computer program, (described in detail in Appendix I), parametric data were generated to determine the minimum blowdown line size permissible consistent with compartmental oil retention. Incorporating the seal environmental pressures and temperatures from FX205-21 into the simulation model provided realistic transient compartment seal conditions.

Figures H-1, H-2, and H-3 illustrate the compartment pressure transients during decel (from intermediate to idle thrust) for the numbers 1, 4, and 5 bearing compartments, respectively, utilizing carbon face type air seals. To properly retain the lubrication system oil within the confines of the bearing compartment, the following minimum blowdown line sizes were determined:

Compartment Number	*Minimum Size* Blowdown Line (ID—Inches)*
1	0.43
4	0.60
5	0.50

*Note: Applicable to Compartments Utilizing Carbon Face Air Seals

Figure H-4 illustrates the air leakage across the carbon face seals at intermediate power for the numbers 1, 4, and 5 compartments as a function of blowdown line size. A composite plot of the compartmental leakage indicates approximately 60 lb/hour total flow for the three blowdown compartments.

A similar set of parametric results was generated utilizing labyrinth seals instead of carbon face seals. Figures H-5, H-6, and H-7 illustrate the compartment pressure transients during decel for the numbers 1, 4, and 5 compartments, respectively, utilizing labyrinth type air seals.

The resulting minimum blowdown line sizes required to prevent compartmental oil loss are tabulated below:

Compartment Number	*Minimum Size** Blowdown Line (ID—Inches)*
1	0.60
4	1.00
5	0.75

**Note: Applicable to Compartments Utilizing Labyrinth Air Seals

Figure H-8 illustrates the air leakage across the labyrinth seals at intermediate power for the numbers 1, 4, and 5 compartments as a function of blowdown line size. A composite plot of the compartmental leakage indicates approximately 925 lb/hour total flow (at intermediate power) for the three blowdown compartments. This is more than 15 times the composite air leakage of the carbon face seal system.

2. SCAVENGE BREATHER SYSTEM RESULTS

In subsequent evaluations it was found that significant rating advantages could be achieved by eliminating the carbon face seal assemblies in favor of the labyrinth type air seals. These advantages resulted from improvements in maintenance man-hours when carbon face seals were replaced with labyrinth seals. Unfortunately, the labyrinth seals resulted in excessive air leakage (≈925 lb/hour intermediate power) when utilized in a scavenge blowdown system. An analysis was conducted in which labyrinth seals were evaluated in conjunction with scavenge pumps sized to minimize air leakage and to prevent compartmental oil loss during engine decel. The analysis (described in detail in Appendix J) indicated that this scavenge breather system was practical from both an air leakage and an oil retention standpoint.

Figure H-9 illustrates the transient seal pressure drop during engine decel for the numbers 1, 4, and 5 compartmental air seals. No reversals in compartment seal ΔP's occur.

Figure H-10 illustrates the transient seal air leakage occurring during engine decel for the numbers 1, 4, and 5 bearing compartments. A composite air leakage of approximately 163 lb/hour (intermediate power) provided a competitive performance comparison with the carbon type face seals making this system a practical alternative to the blowdown scavenge system previously evaluated.

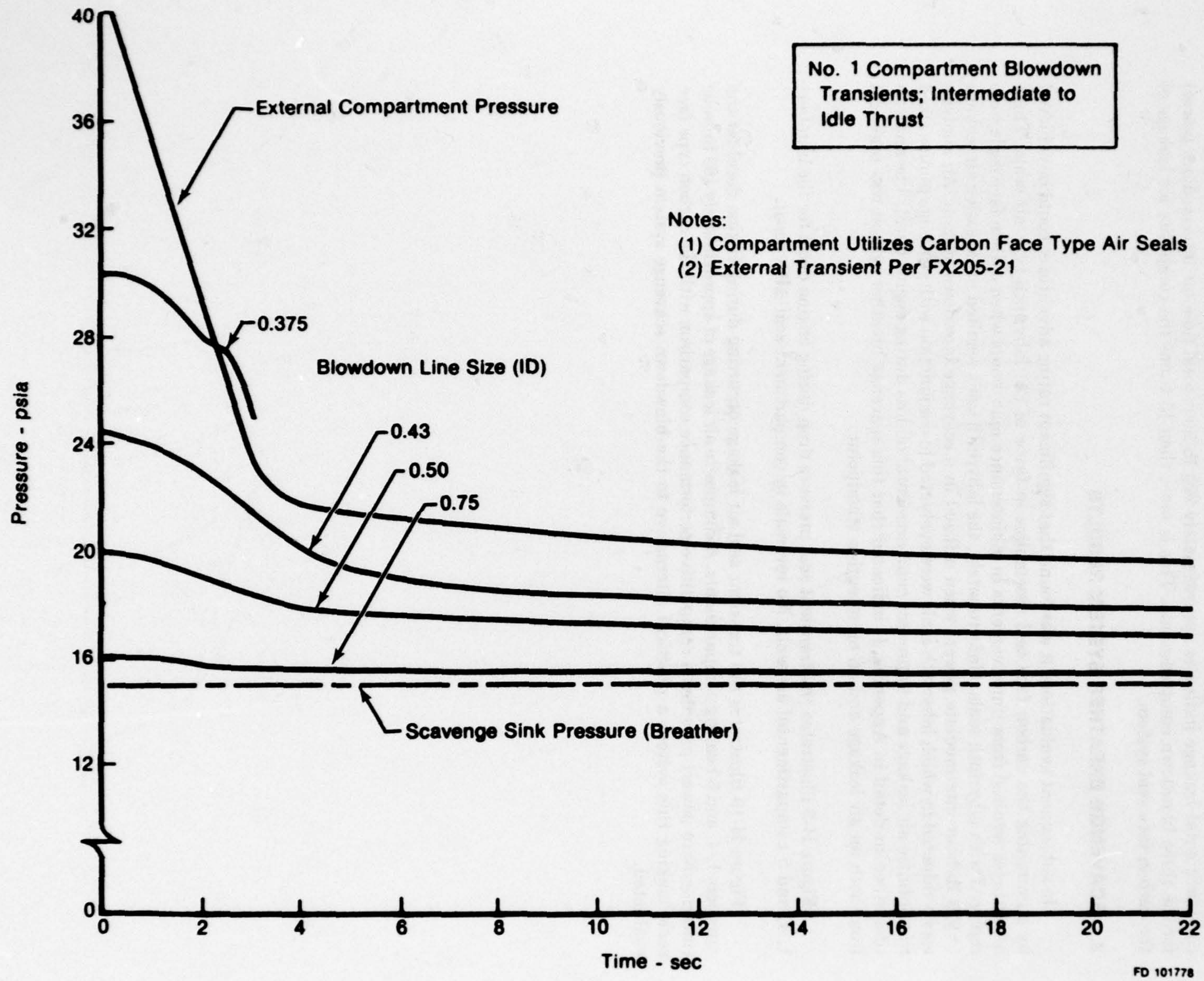


Figure H-1. No. 1 Compartment Blowdown Transient Utilizing Carbon Face Seals

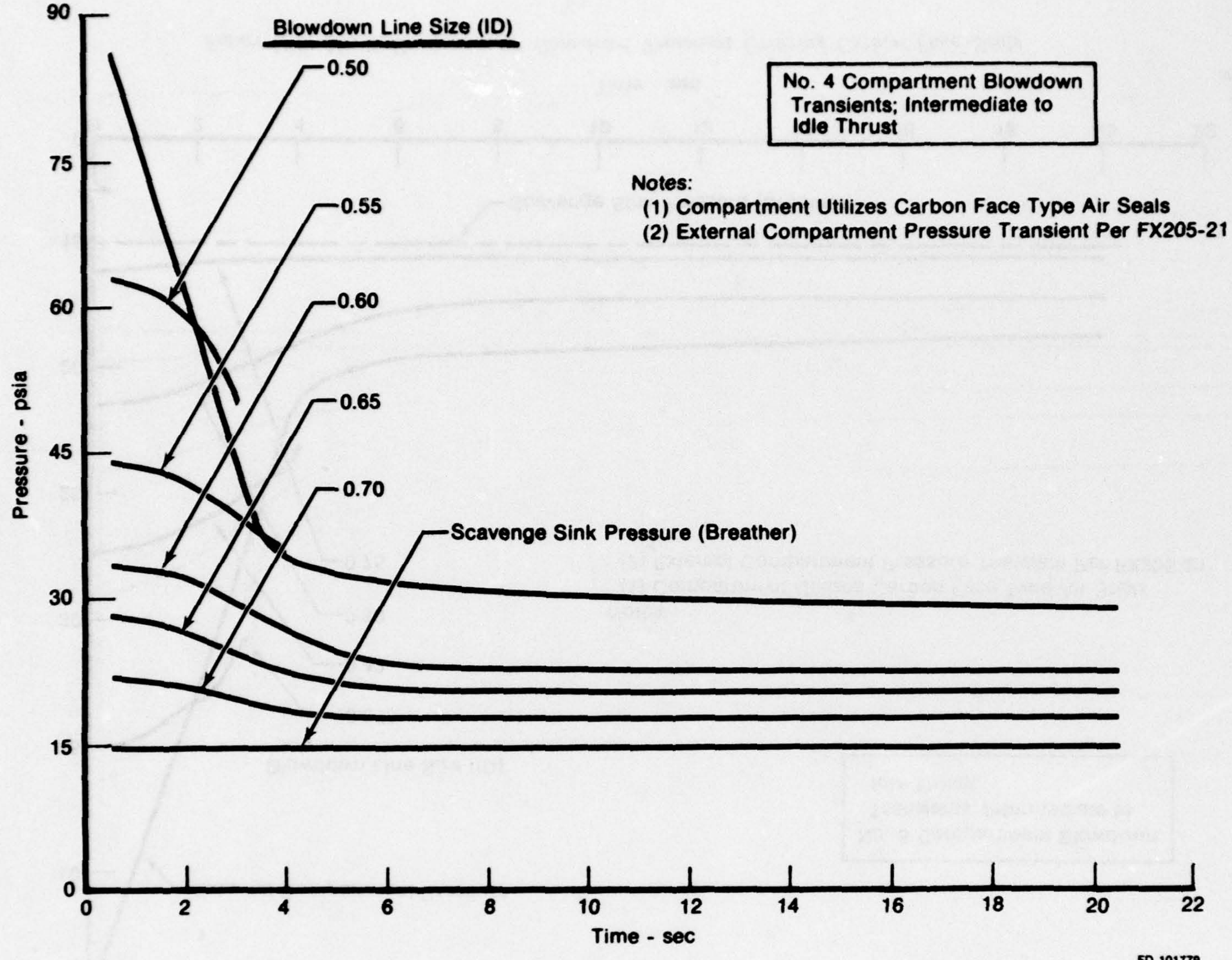


Figure H-2. No. 4 Compartment Blowdown Transient Utilizing Carbon Face Seals

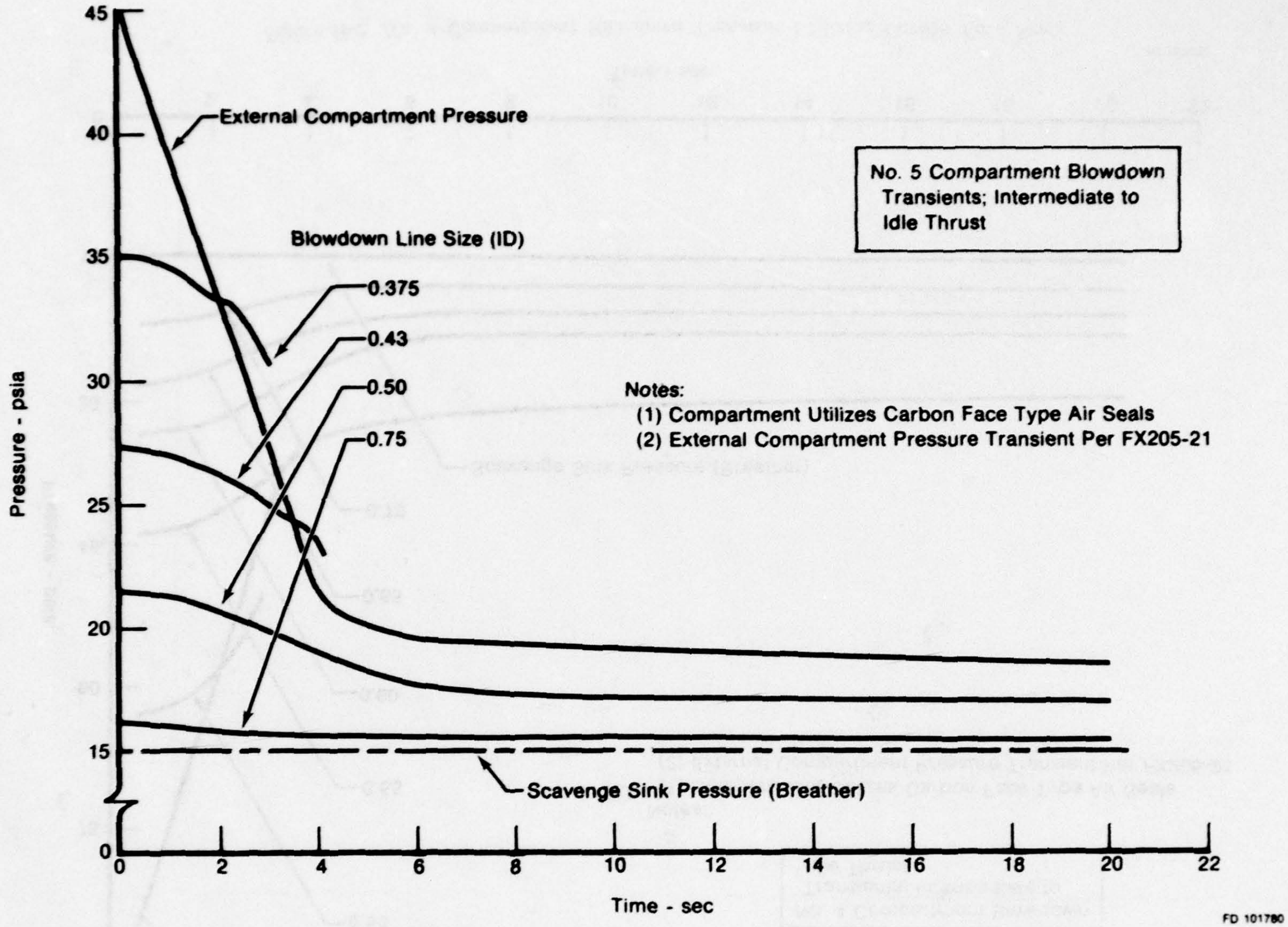


Figure H-3. No. 5 Compartment Blowdown Transient Utilizing Carbon Face Seals

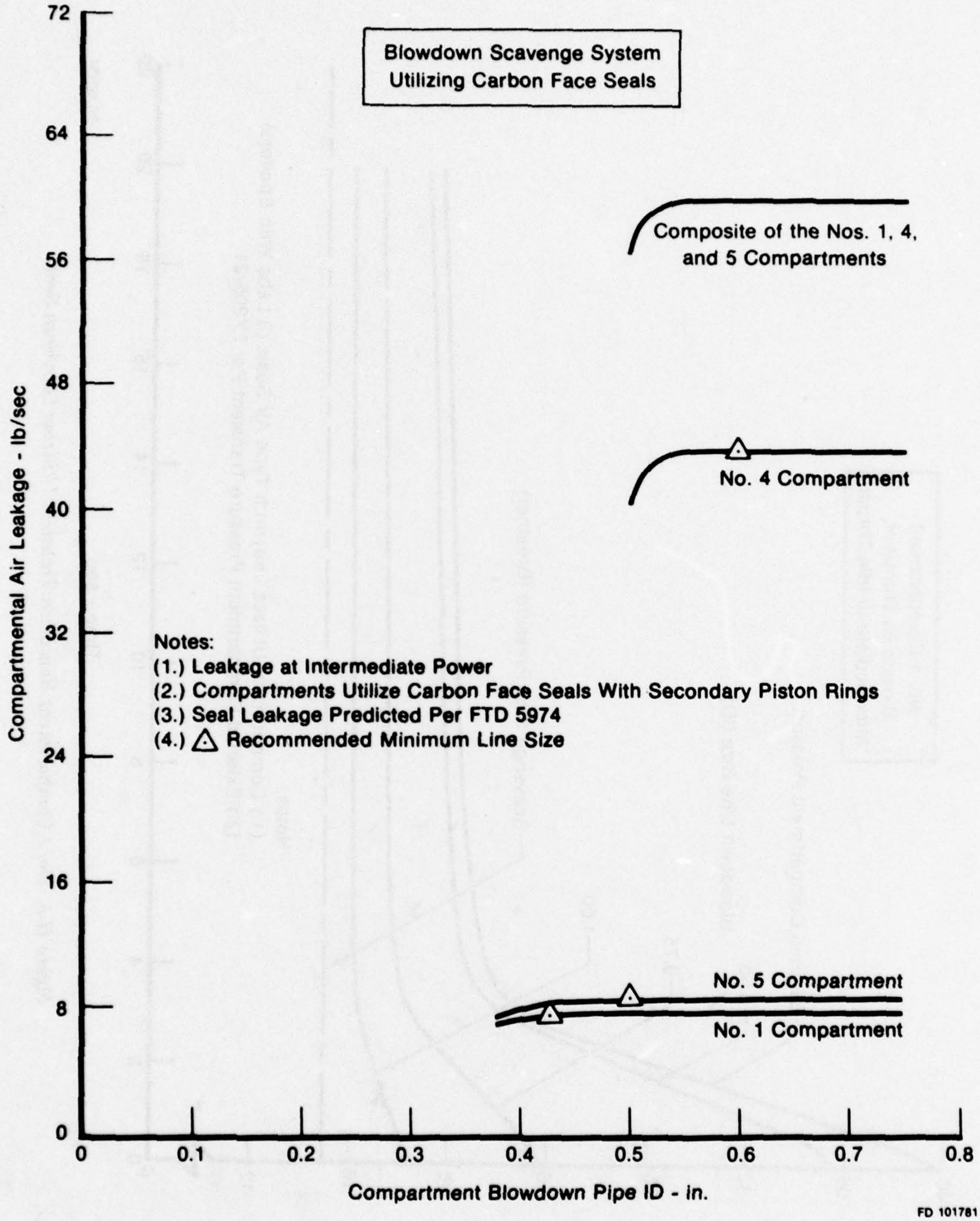


Figure H-4. Blowdown Scavenge Air Leakage Utilizing Carbon Face Seals

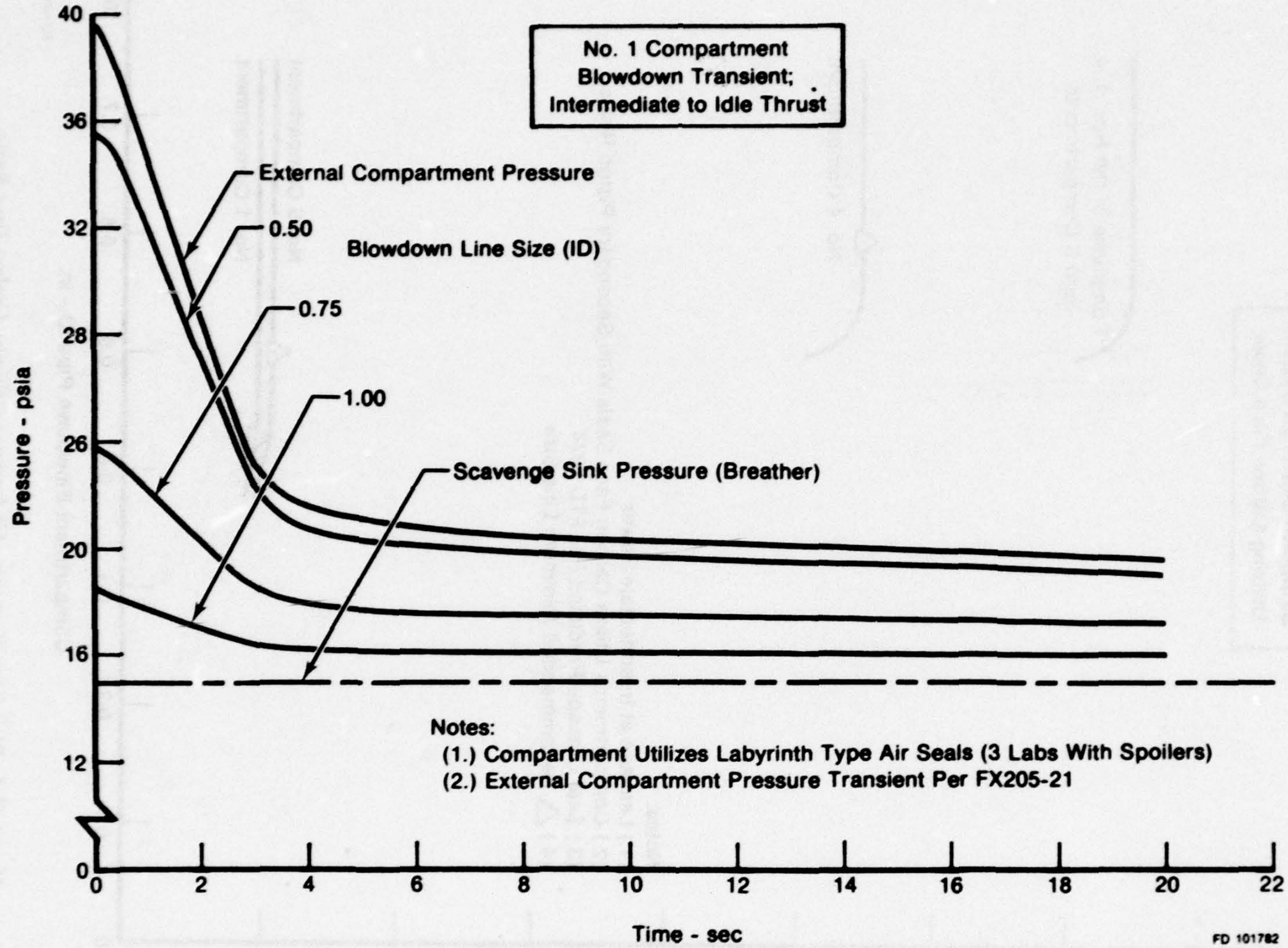


Figure H-5. No. 1 Compartment Blowdown Transient Utilizing Labyrinth Seals

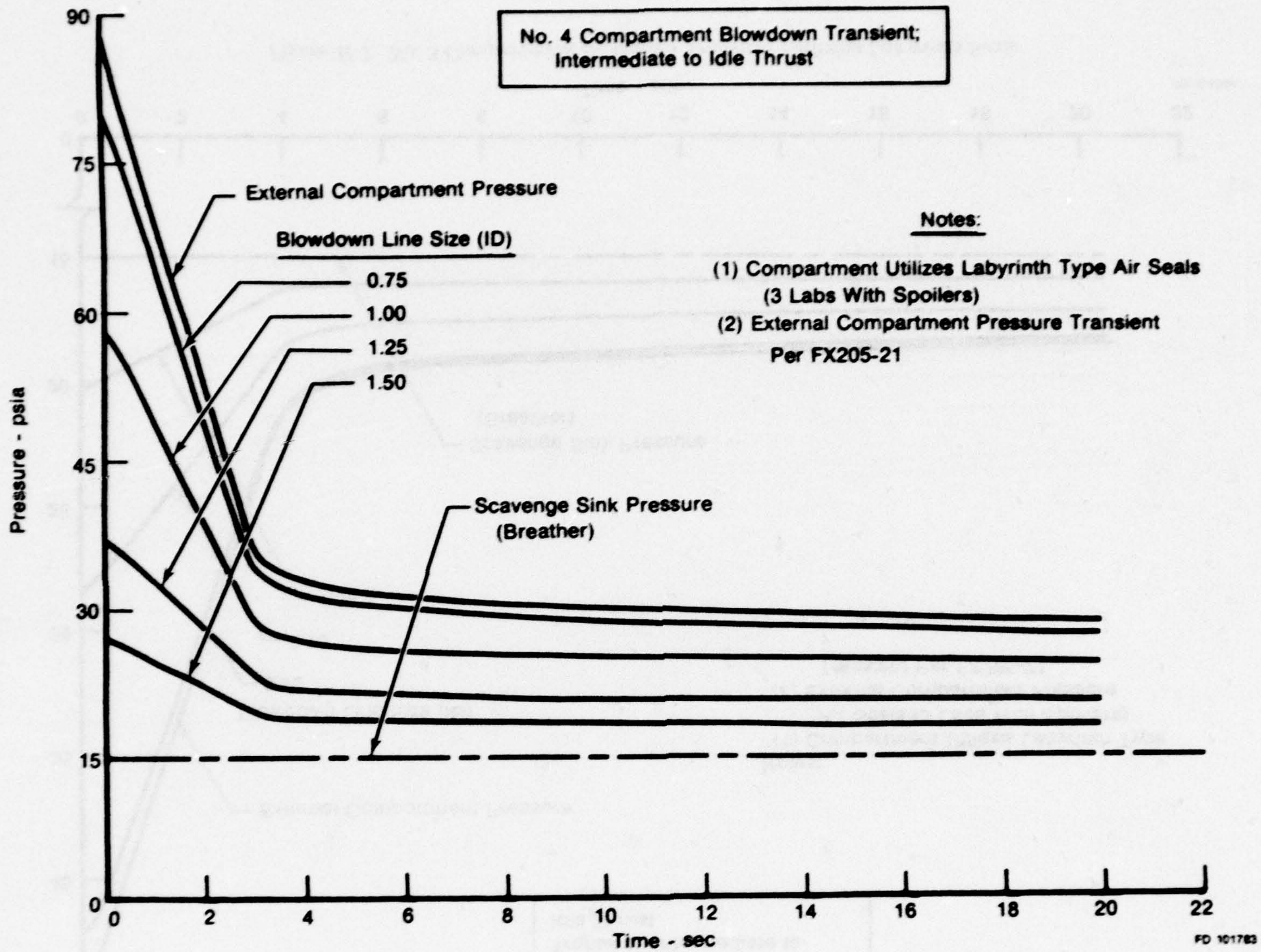


Figure H-6. No. 4 Compartment Blowdown Transient Utilizing Labyrinth Seals

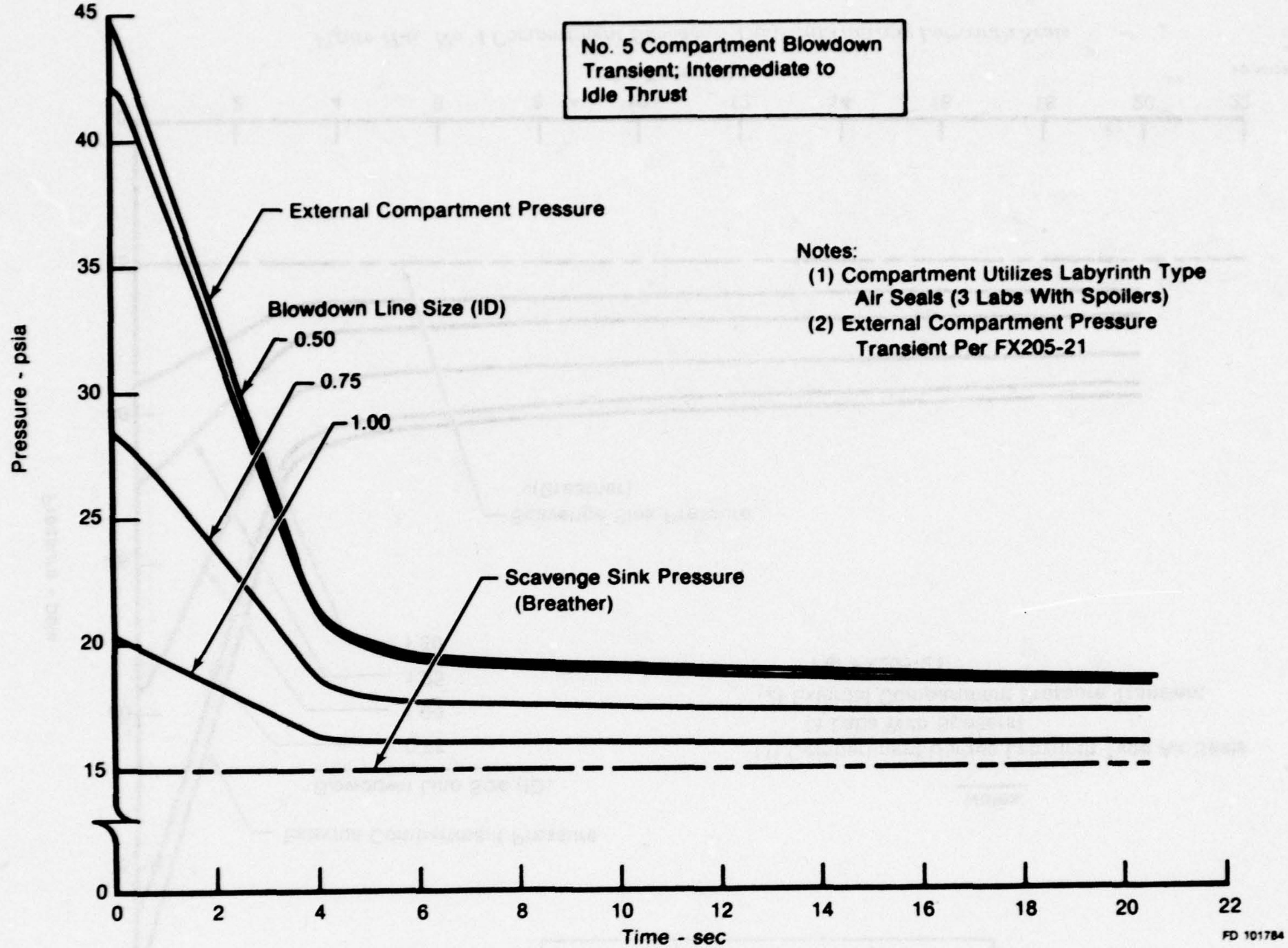


Figure H-7. No. 5 Compartment Blowdown Transient Utilizing Labyrinth Seals

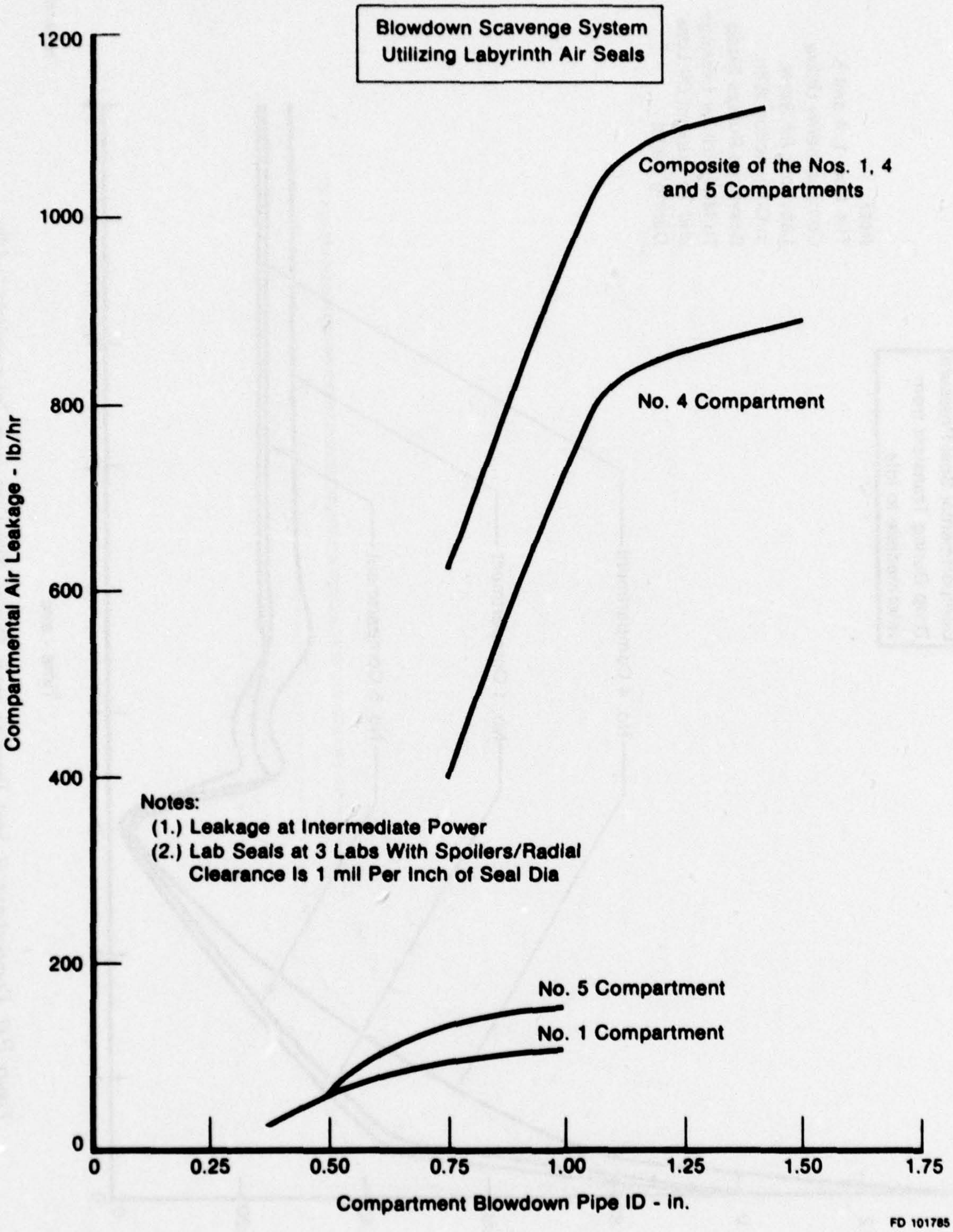


Figure H-8. Blowdown Scavenge Air Leakage Utilizing Labyrinth Airseals

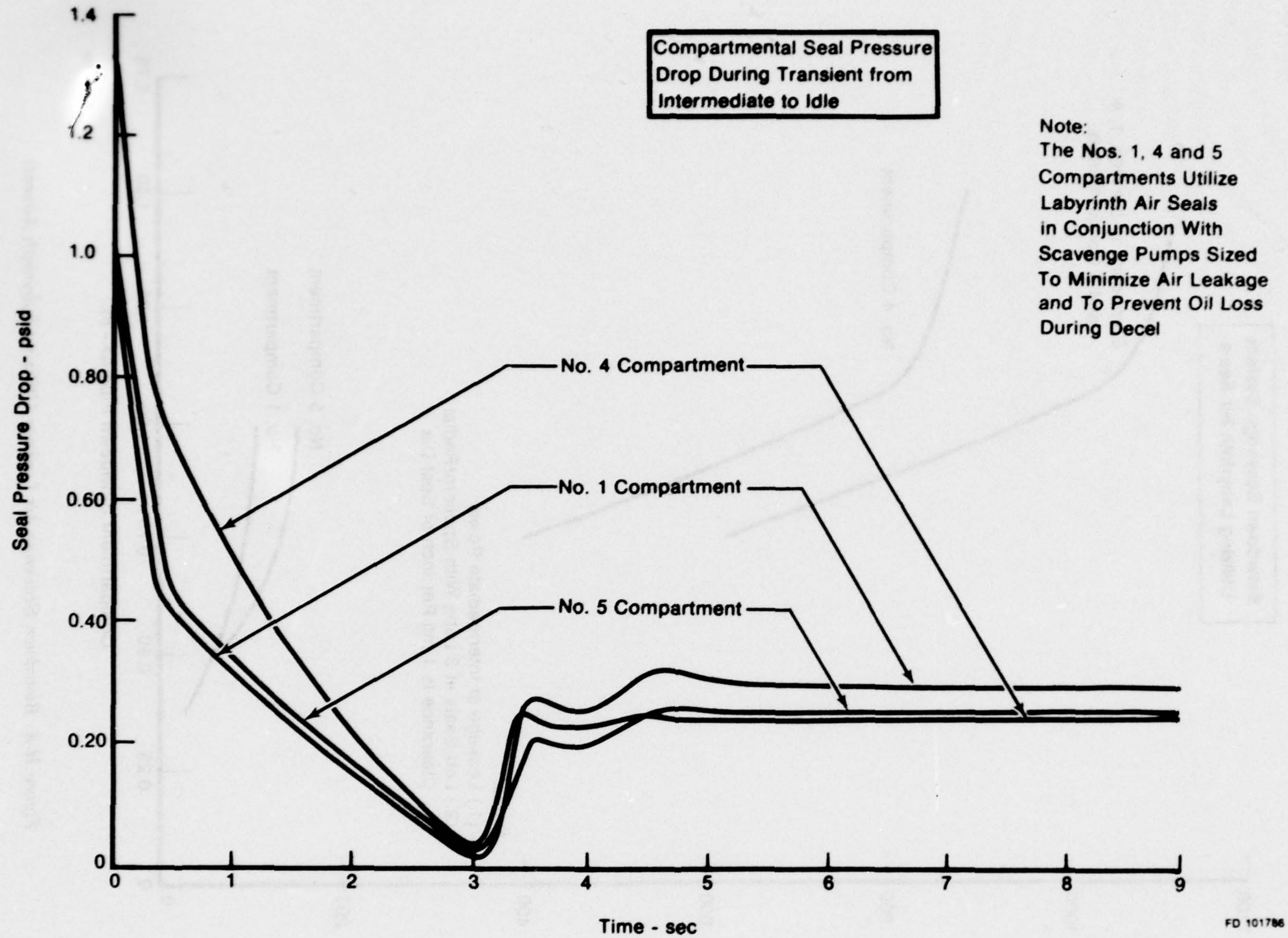


Figure H-9. Compartmental Seal Pressure Drop During Transient from Intermediate to Idle

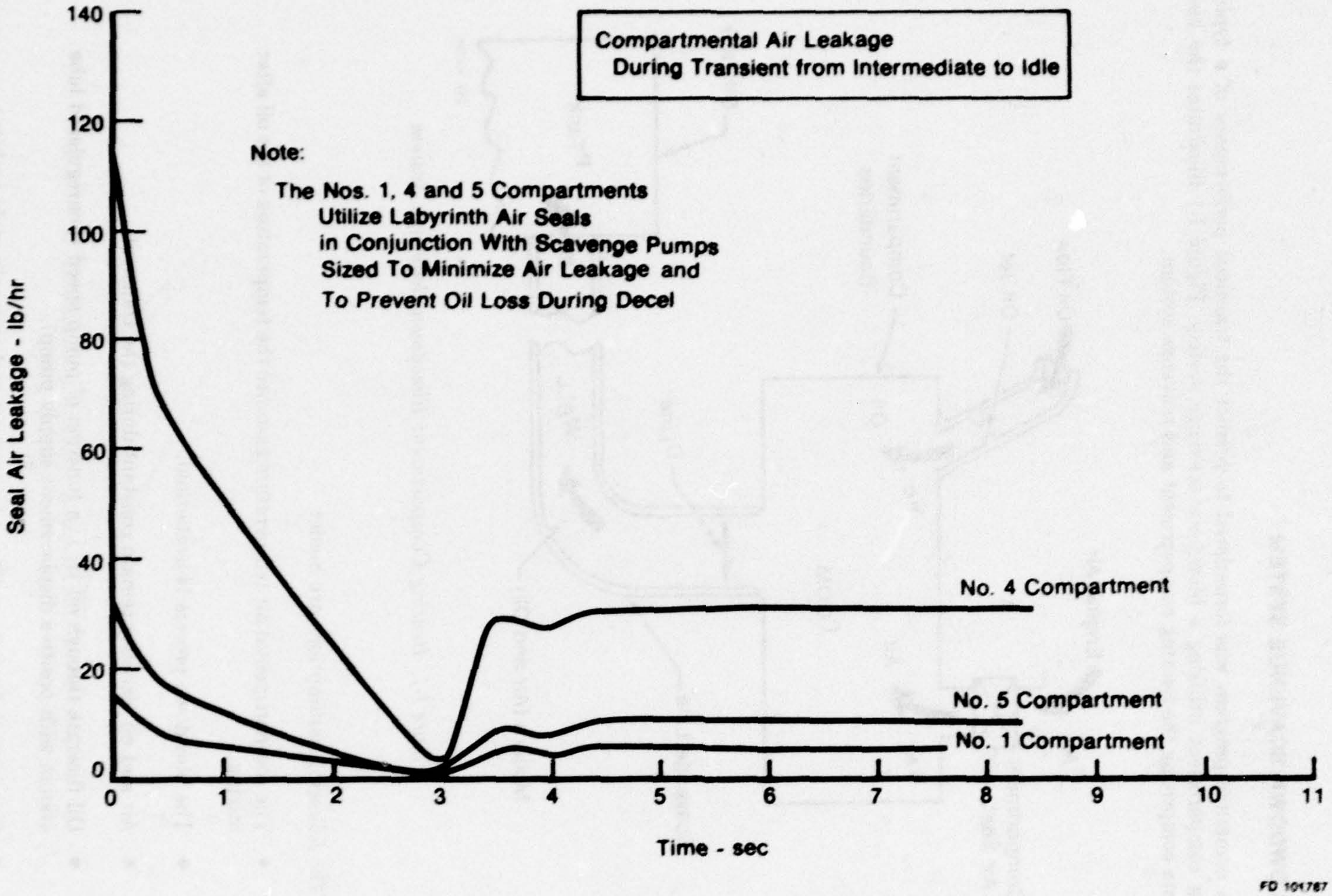


Figure H-10. Compartment Air Leakage During Transient from Intermediate to Idle

APPENDIX I
BLOWDOWN SCAVENGE ANALYSIS

1. BLOWDOWN SCAVENGE SYSTEM

A computer program was formulated to predict the transient performance of a typical bearing compartment utilizing a blowdown scavenge system. Figure I-1 illustrates the basic elements comprising the bearing compartment and scavenge system.

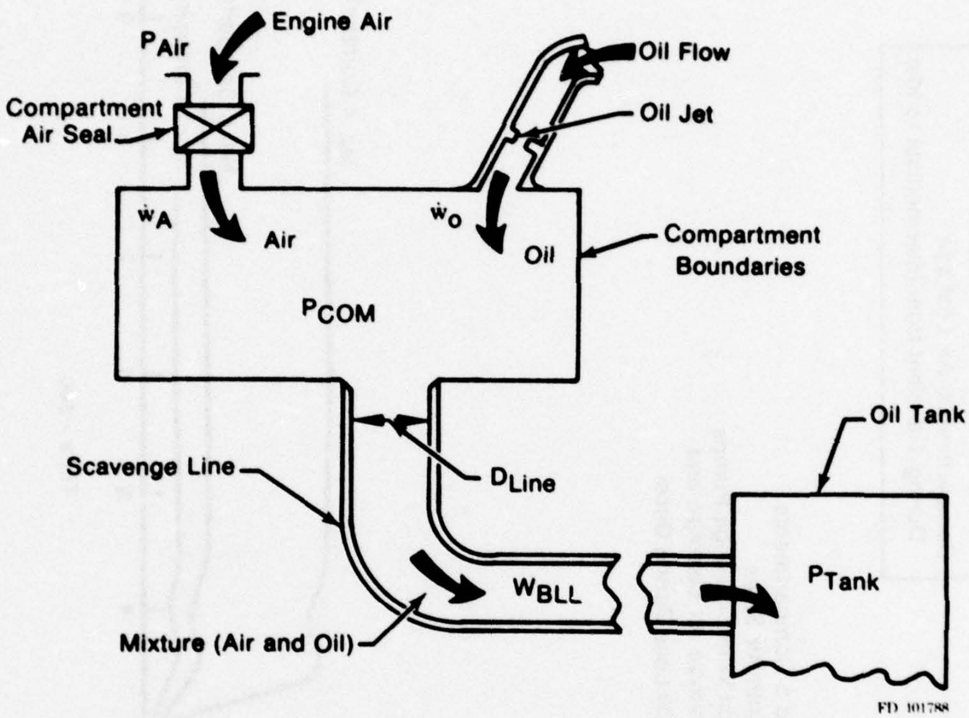


Figure I-1. Bearing Compartment Blowdown Scavenge System

The following assumptions are made:

- The compartmental air temperature assumes the temperature of the oil after mixing.
- The blowdown process is isothermal.
- Air and oil viscosity remain constant during the transient.
- Oil flowrate through oil jet is a function of pump speed (nonregulated lube system with positive displacement supply pump).
- Scavenge sink pressure (P_{tank}) is constant and approximately ambient.

2. DESCRIPTION OF COMPARTMENT PRESSURE

At any point in time the compartment pressure may be described by the perfect Gas Law:

$$(1)\quad P_{com} = \frac{(M_A)(R)(T_{oil})}{V_A}$$

where:

M_A = Mass of air inside compartment, (lb_m)
R = 640.3 in.-lb_f/lb_m-°R; (gas constant)
T_{oil} = Oil temperature, °R
V_A = Volume of air in compartment, in.^{3}
P_{com} = Compartment pressure, psia

The rate of change of compartment pressure with respect to time can be determined by differentiating equation (1):

$$(2)\quad \frac{dP_{com}}{dt} = \dot{P}_{com} = \left(\frac{R\,T_{oil}}{V_A}\right)\dot{M}_A - \left(\frac{M_A\,R\,T_{oil}}{V_A^2}\right)\dot{V}_A$$

Since

$$\dot{M}_A = (W_A - W_{ABLL})$$

where:

W_A = Compartment seal air leakage (lb/sec)
W_{ABLL} = Air flow through blowdown line (lb/sec)

and

$$P_{com} = \frac{M_A\,R\,T_{oil}}{V_A}$$

Then

$$(3)\quad \dot{P}_{com} = [R(T_{oil})/V_A](W_A - W_{ABLL}) - (P_{com}/V_A)(\dot{V}_A)$$

The term $\dot{V}_A$ is the time rate of change of air volume within the compartment and is described by:

$$(4)\quad \dot{V}_A = (1/\rho_o)[W_{oBLL} - W_o]$$

where:

W_{oBLL} = Oil flow through blowdown line (lb/sec)
W_o = Oil jet flow into compartment (lb/sec)
ρ_o = Oil density (lb/in.^{3})

Combining equations (3) and (4) yields:

$$(5)\quad \dot{P}_{com} = [R(T_{oil})/V_A]\,(W_A - W_{ABLL}) - [P_{com}/V_A]\,(1/\rho_o)(W_{oBLL} - W_o)$$

3. DESCRIPTION OF COMPARTMENT AIR LEAKAGE (W_A)

The blowdown scavenge simulation provided an option permitting either carbon face or labyrinth type air seals to be considered.

a. Carbon Face Seals

The airflow through carbon face seals was computed by the general isentropic flow equation:

$$(6)\quad W_A = \frac{(P_{air})(A_{EFF})}{\sqrt{T_{air}}} \left[f(P_{air}/P_{com})\right]$$

where:

$f(P_{air}/P_{com})$ = Isentropic flow parameter
P_{air}, T_{air} = Pressure, temperature outside of compartment seal, psia, °R
A_{EFF} = Seal effective area, in.^{2}

The seal effective area was determined from empirical test data and is computed as follows:

$$(A_{EFF})_{carbon\ face} = (0.000429)(D_{cf})$$

where:

D_{cF} = Diameter of carbon face

$$(A_{EFF})_{piston\ ring} = (0.000143)(D_{PR})$$

where:

D_{PR} = Diameter of secondary piston ring seal

$$(7)\quad A_{EFF} = \text{Total leakage area} = (A_{EFF})_{carbon\ face} + (A_{EFF})_{piston\ ring}$$

b. Labyrinth Seals

The airflow through labyrinth type air seals was computed by the use of seal flow parameters determined from test. Figure I-2 illustrates flow parameters for various labyrinth configurations. The flow characteristics shown were incorporated into the program as an available option.

4. DESCRIPTION OF COMPARTMENT OIL FLOW (W_O)

The oil jets were assumed to be supplied by a positive displacement lubrication pump operating in a nonregulated system. The oil jet flowrate was therefore assumed to track pump speed (linearly). The blowdown transient simulation incorporates a pump speed versus time characteristic as input (along with engine pressures and temperatures) and computes the oil jet flowrate at each discrete time increment as a function of this characteristic.

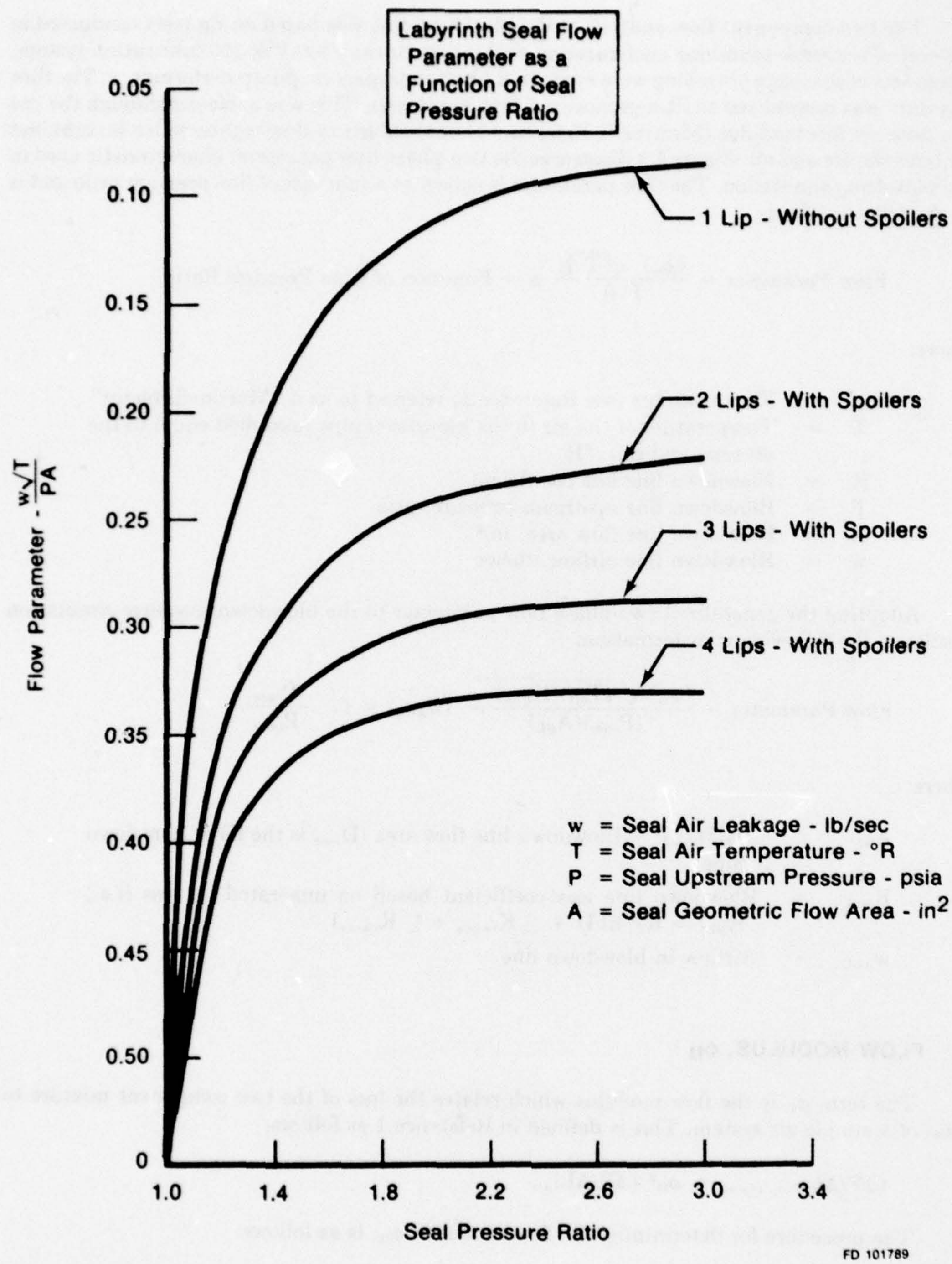


Figure I-2. Labyrinth Seal Flow Parameter as a Function of Seal Pressure Ratio

5. DESCRIPTION OF MIXED (TWO-COMPONENT) FLOW THROUGH BLOWDOWN LINE

The two-component flow analysis of the blowdown line was based on rig tests conducted in support of scavenge plumbing configuration evaluation for the F100-PW-100 lubrication system. Three sets of scavenge plumbing were evaluated for their impact on pump performance. The flow test data was normalized to fit a generalized flow parameter. This was achieved through the use of a pressure loss modulus (Martinelli, Reference 1) applicable to a flow regime which is turbulent for both the air and oil. Figure I-3 illustrates the two-phase flow parameter characteristic used in the blowdown simulation. The flow parameter is shown as a function of line pressure ratio and is in the following form:

$$\text{Flow Parameter} = \frac{(\phi_{tt})\sqrt{T\,K}}{P\,A}\,\dot{w} = \text{Function of Line Pressure Ratio}$$

where:

- ϕ_{tt} = Flow modulus (see Reference 1) referred to as a "Martinelli factor"
- T = Temperature of the air in the blowdown pipe (assumed equal to the oil temperature), °R
- K = Blowdown line loss coefficient
- P = Blowdown line upstream pressure, psia
- A = Blowdown line flow area, in.^{2}
- $\dot{w}$ = Blowdown line airflow, lb/sec.

Adapting the generalized two-phase flow parameter to the blowdown scavenge simulation results in the following transformation:

$$\text{Flow Parameter} = \frac{(\phi_{tt})\sqrt{(T_{oil})(K_{BLL})}}{(P_{com})(A_{BL})}\,(\dot{w}_{ABL}) = f\left(\frac{P_{com}}{P_{tank}}\right)$$

where:

- A_{BL} = $\pi/4(D_{line})^2$ = Blowdown line flow area (D_{line} is the ID of blowdown pipe)
- K_{BLL} = Blowdown line loss coefficient based on unaerated oil loss (i.e., $K_{BLL} = f(\sum 4fl/D + \sum K_{bends} + \sum K_{turns})$
- $\dot{w}_{ABL}$ = Airflow in blowdown line.

6. FLOW MODULUS, ϕ_{tt}

The term ϕ_{tt} is the flow modulus which relates the loss of the two-component mixture to that of a simple air system. This is defined in Reference 1 as follows:

$$(\Delta P/\Delta L)_{two\ phase} = \phi_{tt}^2\,(\Delta P/\Delta L)_{air}$$

The procedure for determining the flow modulus, ϕ_{tt}, is as follows:

Compute

1. Oil to airflow ratio in blowdown line, $\chi_w = (w_{oil})_{BLL}/(\dot{w}_{ABL})$

Two-Phase Flow Parameter as a Function of Pressure Ratio (Applies to a Flow Regime in Which the Air and Oil Flow Are Both Turbulent)

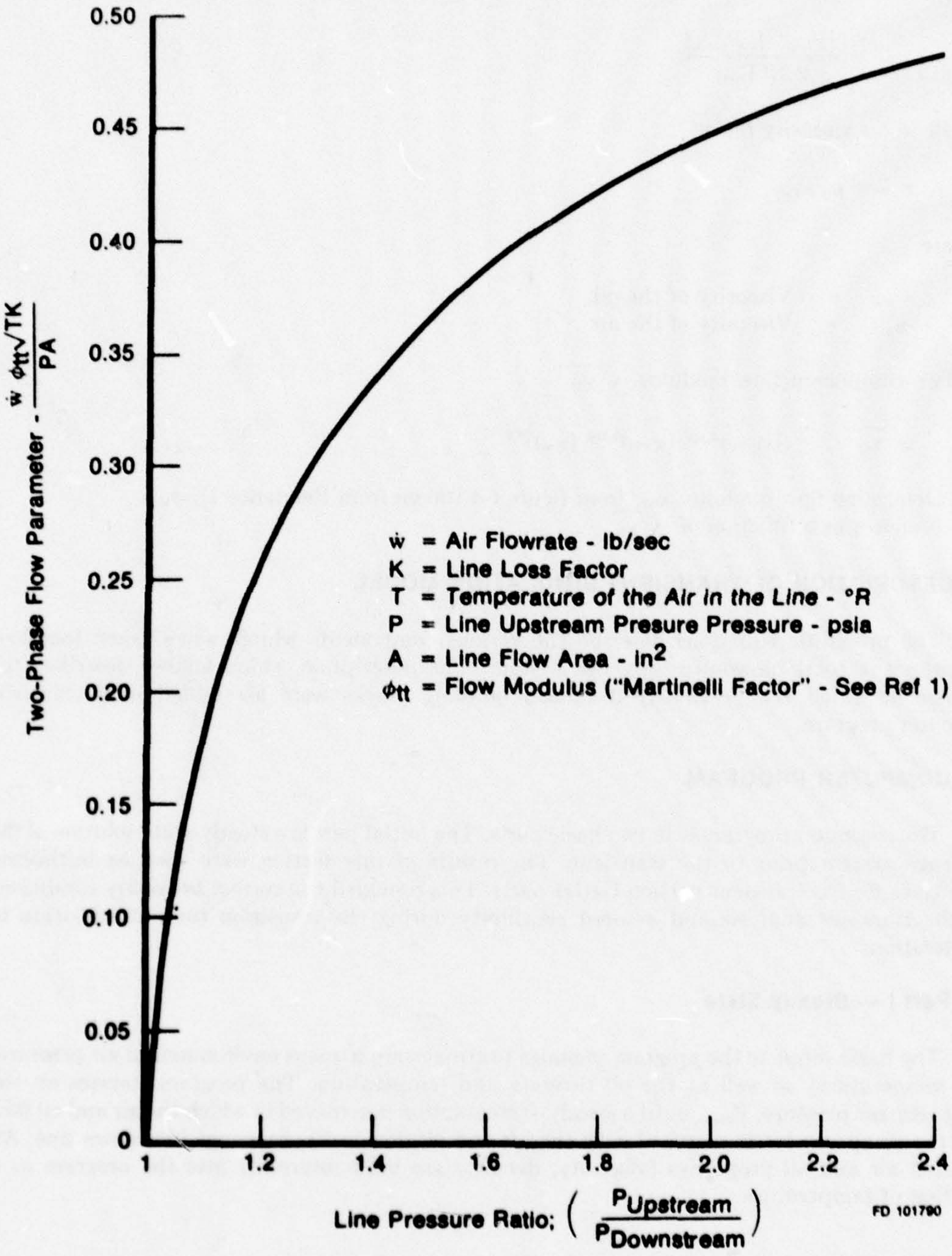


Figure I-3. Two-Phase Flow Parameter as a Function of Pressure Ratio

2. Air to oil density ratio in blowdown line, $\chi_{TD} = \rho_{air}/\rho_o$

where:

$$\rho_{air} = \frac{[P_{com} + P_{tank}]}{2\ R(T_{oil})}$$

3. Oil to air viscosity ratio:

$$\chi_{TV} = \mu_o/\mu_{air}$$

where:

μ_o = Viscosity of the oil
μ_{air} = Viscosity of the air

4. Two-component flow modulus, $\sqrt{\chi_{tt}}$

$$\sqrt{\chi_{tt}} = [(\chi_{TV})^{0.111} (\chi_{TD})^{0.555} (\chi_w)]^{0.5}$$

5. Determine flow modulus, ϕ_{tt}; from figure I-4 (taken from Reference 1). ϕ_{tt} is shown as a function of $\sqrt{\chi_{tt}}$.

7. DESCRIPTION OF TRANSIENT SIMULATION MODEL

The preceding equations describe the various components which, when taken together, comprise the total blowdown scavenge system. The description which follows describes the manner in which the previously discussed building blocks were assembled in a transient computer program.

8. COMPUTER PROGRAM

The computer program is in two basic parts. The initial part is a steady-state solution of the scavenge system prior to the transient. The results of this section were used as initialized conditions for the transient portion (latter part). This provided the correct boundary conditions for the transient analysis and assured continuity during the transition from steady-state to deceleration.

a. Part I — Steady-State

The basic input to the program includes bearing compartment environmental air pressures and temperatures as well as the oil flowrate and temperature. The program iterates on the compartment pressure, P_{com}, until a steady-state solution is achieved in which the air and oil flow into the compartment is matched with the air and oil flow in the scavenge blowdown line. All required air and oil properties (viscosity, density) are built internally into the program as a function of temperature.

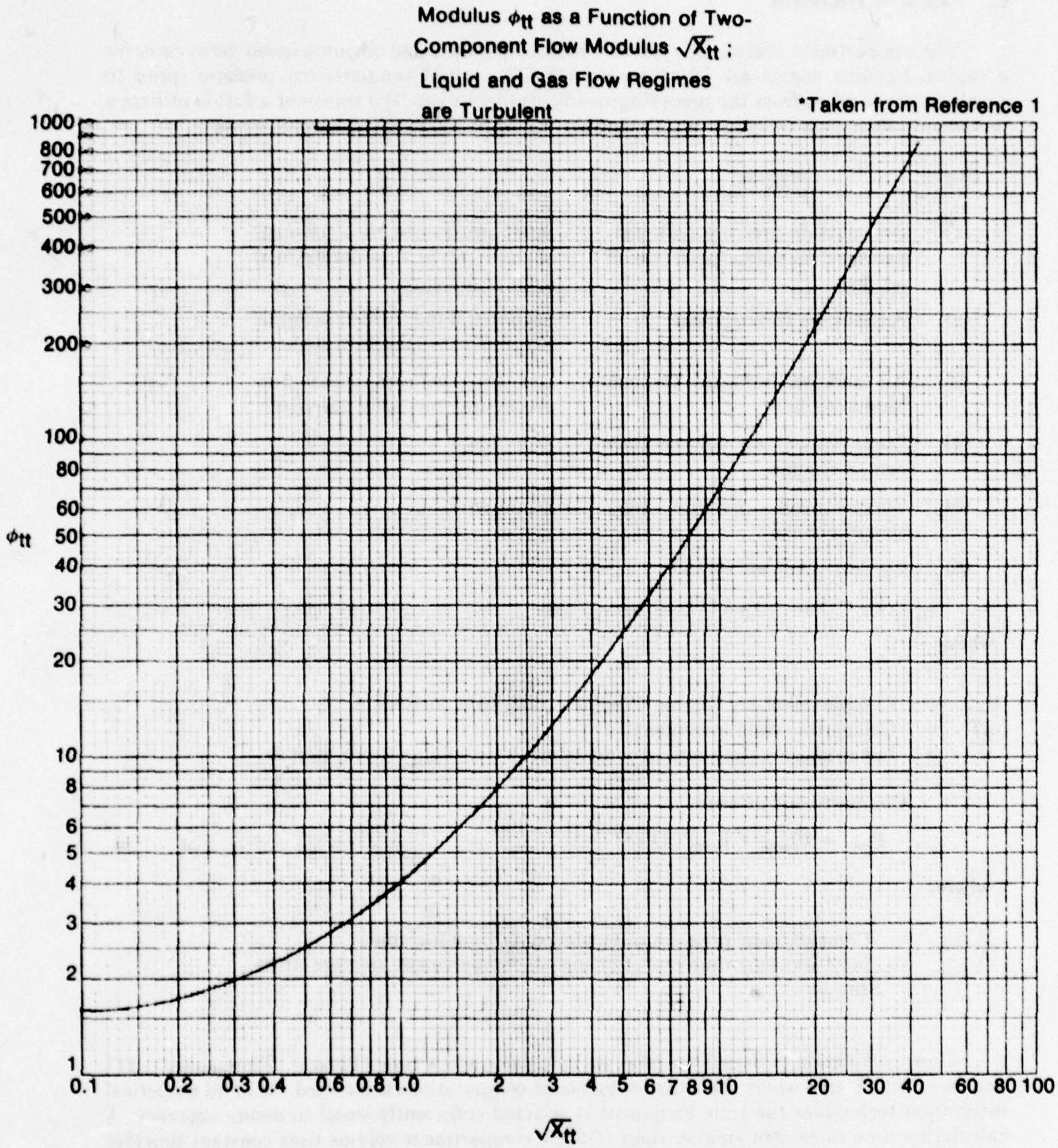


Figure I-4. Two Component Flow Modulus

b. Part II — Transient

The compartment seal outside pressure and temperature and oil pump speed decay rates for a typical baseline engine are input programed. The initial compartment pressure (prior to deceleration) is taken from the preceding steady-state solution. The transient solution utilizes a numerical integration technique employing the following sequence of computations:

Step	*Compute*	*Procedure*
(1)	Seal outside pressure and temperature; oil pump speed and oil jet flow	Input programed as a function of time; oil jet flow determined from pump speed
(2)	Air leakage through seal	Equation 6 (for carbon seals) or Figure I-2 for labyrinth seals
(3)	Air and oil flowrates through blowdown line	Use the two-component flow procedure previously described
(4)	Compartment air volume time rate of change	Equation 4
(5)	Compartment pressure time rate of change	Equation 5
(6)	Compartment air volume	

$$V_A = V_{AP} + \dot{V}_A \, (dT)$$

where:

V_{AP} = Compartment air volume from preceding time increment
dT = Calculating time increment
$\dot{V}_A$ = Instantaneous time rate of compartment air volume (from Step 4)

(7) Compartment pressure

$$P_{com} = P_{comp} + \dot{P}_{com} \, (dT)$$

where:

V_{comp} = Compartment pressure from preceding time increment
$\dot{P}_{comp}$ = Instantaneous time rate of change of compartment pressure (from Step 5)

Upon completion of Step 7 the program calculates a new time (Time = Time$_{previous}$ dT) and returns to Step 1 where the entire sequence of computations is recycled. As in all numerical integration techniques the time increment is selected sufficiently small to assure accuracy. A calculating time increment smaller than 1/10 the compartment volume time constant provides this assurance.

9. TWO-COMPONENT PRESSURE LOSS CORRELATION

A correlation between the two-component flow model used in the blowdown scavenge simulation and the No. 3/3½ compartment blowdown scavenge system of the JT15D was performed. The JT15D is a production engine (Pratt & Whitney Aircraft of Canada) in a less than 10,000 lb thrust class which utilizes a blowdown scavenge system in the No. 3/3½ compartment. The JT15D blowdown line resistance (from compartment to gearbox) is heavily dominated by a 0.300 diameter section of plumbing designed to provide a compartment backpressure effect to minimize seal air leakage by reducing seal ΔP.

Figure I-5 illustrates the prediced flow characteristics of the blowdown line for steady-state pressure and temperature conditions taken from the JT15D. Superimposed on this figure are the estimated JT15D No. 3/3½ compartment air and oil flows. The two-component flow model predicts approximately 75 percent of the estimated JT15D air leakage at the estimated JT15D oil flow.

No compartment transient pressure data is available from the JT15D for comparison with the transient simulation used in the blowdown scavenge studies.

REFERENCES

1. "Isothermal Pressure Drop for Two-Phase, Two-Component Flow in a Horizontal Pipe," R. C. Martinelli, L. Boelter, T. Taylor, E. Thomsen, and E. Morrin, Transactions of the ASME, February 1944.

2. "Proposed Correlation of Data for Isothermal Two-Phase, Two-Component Flow in Pipes," R. W. Lockhart and R. C. Martinelli, Chemical Engineering Progress, January 1949.

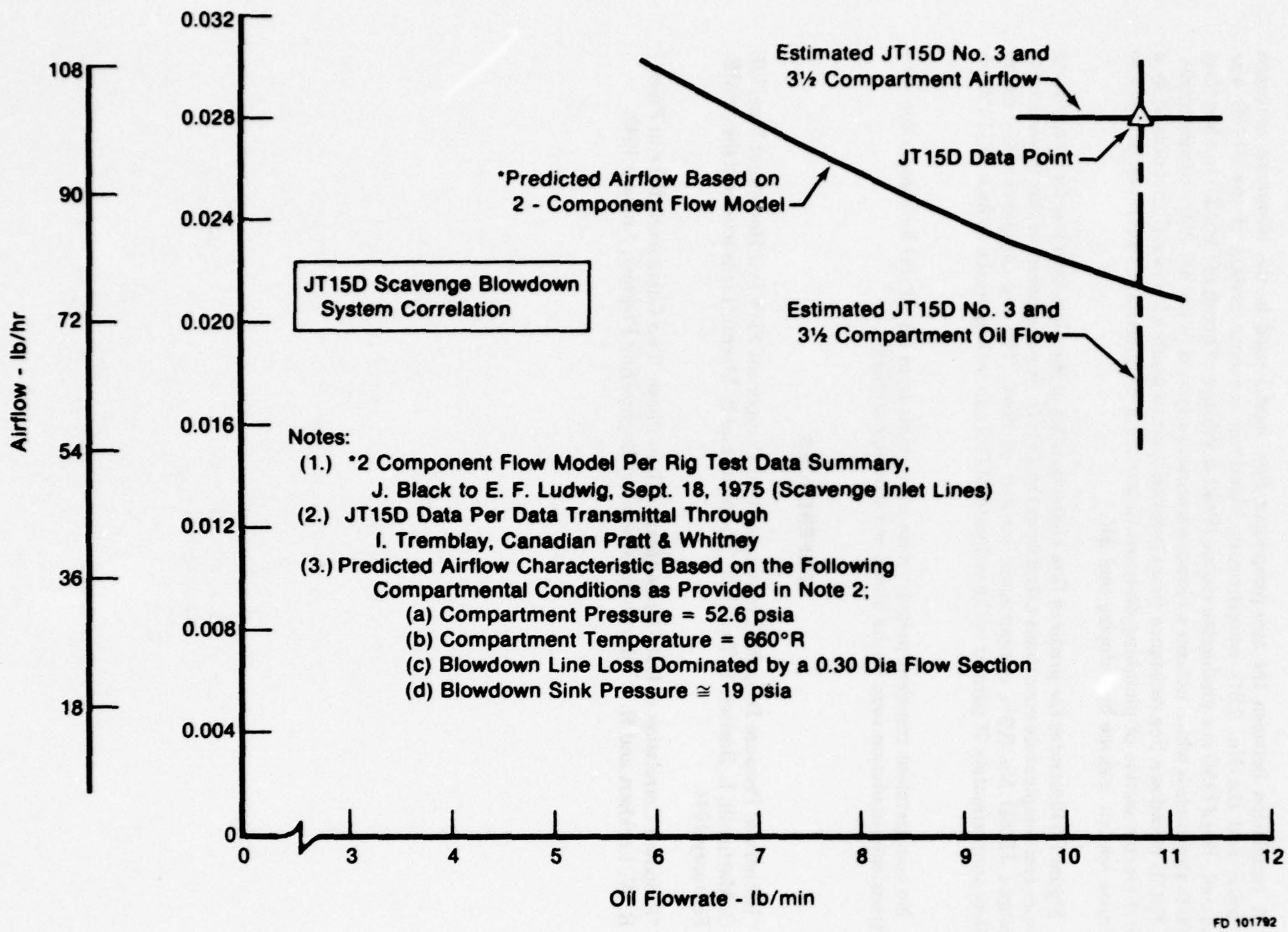


Figure I-5. JT15D Scavenge Blowdown System Correlation

APPENDIX J
SCAVENGE BREATHER ANALYSIS

1. SCAVENGE BREATHER SYSTEM

A computer program was utilized to evaluate the transient bearing compartment pressure characteristics during engine deceleration from intermediate power to idle. This program simulates the scavenge breather system and predicts transient compartment pressures in a manner similar to the blowdown system simulation which is described in detail in Appendix I. The scavenge breather system differs from the blowdown system primarily in the use of a scavenge pump to transfer the compartmental air leakage and oil flow from the compartment to the oil tank. Figure J-1 illustrates the basic elements comprising the scavenge breather system.

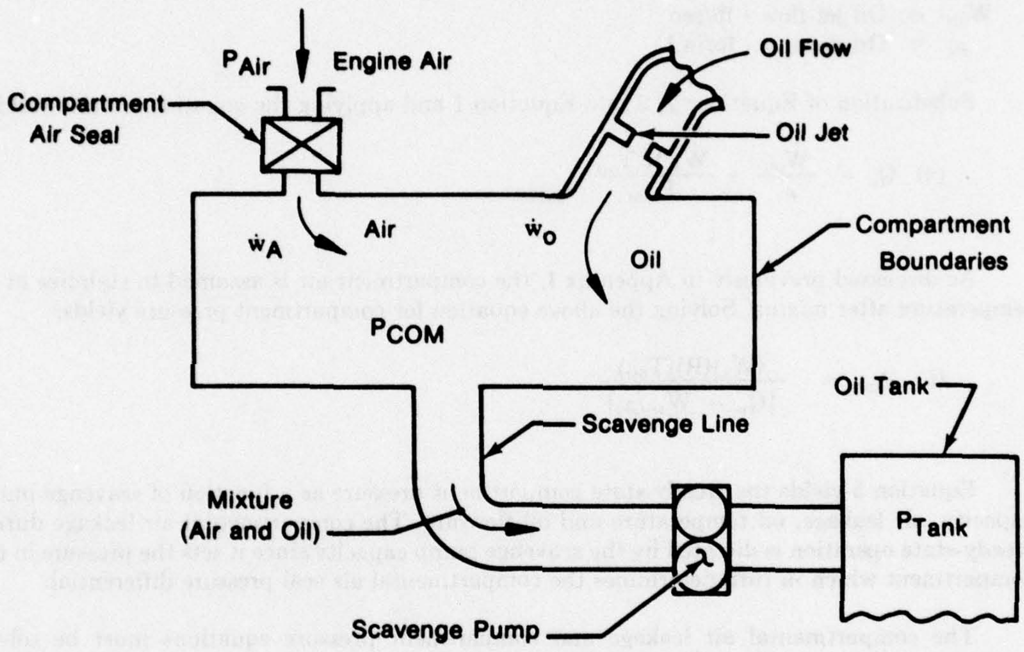


Figure J-1. Bearing Compartment Scavenge Breather System

The scavenge pump is a positive displacement type pump whose volumetric flow capacity is proportional to pump speed. The scavenge pump is sized in excess of the oil flow requirement to assure positive air sealing during engine deceleration.

During steady-state operation, the compartmental air seal leakage and oil jet flow are balanced by the scavenge pump flow output, i.e.,

$$(1) \quad \dot{Q}_P = \dot{Q}_{oil} + \dot{Q}_{air}$$

where:

$\dot{Q}_P$ = Volumetric flow output of scavenge pump - in.^{3}/sec
$\dot{Q}_{oil}$ = Volumetric oil flow into compartment - in.^{3}/sec
$\dot{Q}_a$ = Volumetric air flow into compartment - in.^{3}/sec

(2, 3) Since $\dot{Q}_A = \dot{W}_a/\rho_a$ and $\dot{Q}_{oil} = \dot{W}_{oil}/\rho_o$

where:

$\dot{W}_a$ = Mass airflow leakage across compartment air seals - lb/sec
ρ_a = Compartment air density - lb/in.^{3}
$\dot{W}_{oil}$ = Oil jet flow - lb/sec
ρ_o = Oil density - lb/in.^{3}

Substitution of Equations 2, 3 into Equation 1 and applying the equation of state yields:

$$(4)\quad \dot{Q}_p = \frac{\dot{W}_{oil}}{\rho_o} + \frac{\dot{W}_a R T_{oil}}{P_{com}}$$

As discussed previously in Appendix I, the compartment air is assumed to stabilize at oil temperature after mixing. Solving the above equation for compartment pressure yields:

$$(5)\quad P_{com} = \frac{(\dot{W}_a)(R)(T_{oil})}{[\dot{Q}_p - \dot{W}_{oil}/\rho_o]}$$

Equation 5 yields the steady-state compartment pressure as a function of scavenge pump capacity, air leakage, oil temperature and oil flowrate. The compartmental air leakage during steady-state operation is dictated by the scavenge pump capacity since it sets the pressure in the compartment which in turn determines the compartmental air seal pressure differential.

The compartmental air leakage and compartment pressure equations must be solved simultaneously for a unique solution to exist;

$$(6)\quad \dot{W}_A = f(P_{air}, P_{com}, A_{EFF}, T_{air})$$

(See Appendix I for detail seal flow model.)

The computer program iterates on compartment pressure (P_{com}) until the air leakage in Equations 5 and 6 balance.

2. TRANSIENT SIMULATION

Once a steady-state solution is achieved the results are used as initialized conditions prior to the engine deceleration. The transient pressure analysis is the same as previously discussed in Appendix I except for the time rate of change of air mass in the compartment. Accounting for the scavenge pump airflow and eliminating the blowdown line yields the following:

$$\dot{M}_A = \dot{W}_A - \dot{W}_{AP}$$

where:

$\dot{W}_{AP}$ = Air flowrate through scavenge pump - lb/sec

since:

$$\dot{W}_{AP} = \rho_a \dot{Q}_a = P_{com}/R(T_{oil}) [\dot{Q}_P - \dot{W}_{oil}/\rho_o]$$

Therefore:

$$(7) \quad \dot{M}_a = \dot{W}_A - (P_{com})/(R)(T_{oil})[\dot{Q}_P - \dot{W}_{oil}/\rho_o]$$

The compartment pressure is computed during the transient by integrating the compartment air mass and computing the pressure time derivative in discrete time increments as discussed in Appendix I. Compartment seal outside pressure and temperature and oil supply/scavenge pump speed decay rates for a typical baseline engine (for an engine deceleration) are input programed. The scavenge pump flow capacity is computed from the rotor speed at each time increment in the transient along with all the other parameters.

Comments made in Appendix I relative to the selection of the magnitude of the time increment applies equally well here.

APPENDIX K
OIL PUMP DESIGN

OIL PUMP GEAR STRESSES:

known:

hp	=	4.0 or, 2 hp/stage
Gear pitch dia.	=	0.5625 in.
No. Teeth	=	9
Diametrical Pitch	=	16
Pressure Angle	=	28°
X Factor	=	0.033
Hertz Stress Allowable	=	100,000 psi

REQUIRED FACE WIDTH

$$F = \frac{0.7 \times E \times W \times (M_G+1)}{\sin 2\phi(d)\,(Sc)^2\,(M_G)}$$

$$W = \text{Tangential tooth load} = \frac{2T}{D} = \frac{2 \times 12.6}{0.5625} = 44.8 \text{ lb};$$

$$T = \frac{63{,}000 \times 2}{10{,}000} = 12.6 \text{ in.} - \text{lb}$$

M_G = gear ratio = 1:1
d = pitch dia.
Sc = Hertz stress allowable = 100,000 psi
ϕ = pressure angle = 28°
F = Face width

$$F = \frac{0.7 \times E \times 44.8 \times 2}{\sin\left[(2)(28°)\right](0.5625)(100{,}000)^2} = 0.403$$

Actual F = 1.674

$$(\text{Actual Hertz Stress})^2 = \frac{0.403}{1.674}(100{,}000)^2 = 49{,}065 \text{ psi}$$

$$\therefore SF = \frac{100{,}000}{49{,}065} = 2.038$$

Now determine cyclic bending stress for gear teeth.

1. Find *Dynamic Tooth Load:*

$$Wd = \frac{0.05V\ [FC+W]}{0.05V + [FC+W]^{1/2}} + W$$

V = pitch line velocity ft/min = 1473 ft/min
F = face width = 1.674
C = error factor = 950
W = tangential load (LB) = 44.8 LB

$$Wd = \frac{(0.05)\ (1473)\ [\ 1.674\ (950) + 44.8]}{0.05\ (1473) + [\ 1.674\ (950) + 44.8]^{1/2}} + 44.8$$

Wd = 1101 LB

2. Now calculate *allowable tooth load:*

$$We = \frac{(0.667)\ (S_B)\ (F)\ (X)}{K}$$

We = allowable load
S_B = bending stress (cyclic) = 63,000 psi
F = *face width*
X = tooth factor determined from layout = 0.033
K = stress concentration factor = 1.5

$$We = \frac{(0.667)\ (63,000)\ (1.674)\ (0.033)}{1.5} = 1547$$

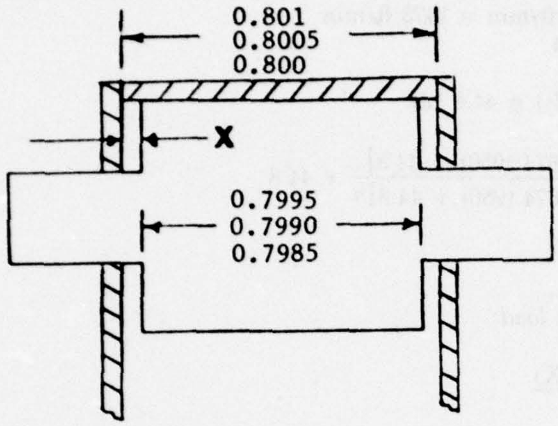
$$SF = \frac{1547}{1101} = 1.405$$

COMPARTMENTAL LUBRICATION SYSTEM OIL PUMP DESIGN

1. Objective

Scale the ST-9 oil pump to F100 oil flows using ST-9 rig data and calculated leakage flows.

2. Calculation of Running (300°F) End Plate Clearance for ST-9



	Cold Clearances		
	Max	*Nominal*	*Min*
	0.8010	0.8005	0.8000
	−0.7985	−0.7990	−0.7995
2X =	0.0025	0.0015	0.0005

a. Gear Length Growth at 300°F

$\Delta l = \alpha L \Delta T$

For AMS-6470 or AMS-6260, $\alpha = 6.6 \times 10^{-6}$ in./in./°F (See Figure K-1) at (300°F)

$\Delta l = (6.6 \times 10^{-6}$ in./in./°F) (0.7995) (300 − 68) = 0.0012242 = 0.00122 in.
(0.7990) 0.0012234
(0.7985) 0.0012227

b. Shell Length Growth at 300°F

For AMS 4120 at 300°F $\alpha = 12.98 \times 10^{-6}$ in./in./°F

$\Delta l = (12.98 \times 10^{-6}$ in./in./°F) (0.801) (300 − 68) = 0.002412
(0.8005) 0.002411
(0.800) 0.002409

Lengths at 300°F

Gear			Shell		
0.7995	0.7990	0.7985	0.8010	0.8005	0.8000
0.0012	0.0012	0.0012	0.0024	0.0024	0.0024
0.8007	0.8002	0.7997	0.8034	0.8029	0.8024

	Running Clearance		
	Max	*Nominal*	*Min*
	0.8034	0.8029	0.8024
	−0.7997	−0.8002	−0.8007
2X =	0.0037	0.0027	0.0017

c. ***Using Actual Pump Measurements***

Gear Length (driven) 0.7983 (driver) 0.7981

average = 0.7982

Housing Length = 0.8012

clearance = 0.8012 − 0.7982 = 0.0030 in.

Gear Growth

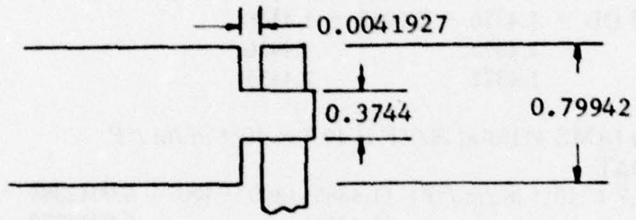
$\Delta \ell = \alpha L \Delta T = (6.6 \times 10^{-6}\ \text{in./in./°F})\ (0.7982)\ (300 - 68) = 0.00122$ in.
Running Length = 0.7982 + 0.0012 = 79942

Shell Growth

$\Delta \ell = \alpha L \Delta T = (12.98 \times 10^{-6}\ \text{in./in./°F})\ (0.8012)\ (300 - 68) = 0.0024127$
Running Length = 0.8012 + 0.0024 = 0.80361

Running Clearance = 0.80361 − 0.79942 = 0.00419

End Plate Area Calculation



Shaft Growth = $(6.6 \times 10^{-6}$ in./in./°F) (0.3738) (300 − 68) = 0.00057
Diameter = 0.3738 + 0.00057 = 0.3744

Side plate total length cold = (2) (gear pitch radius) + (2) (gear outside radius)

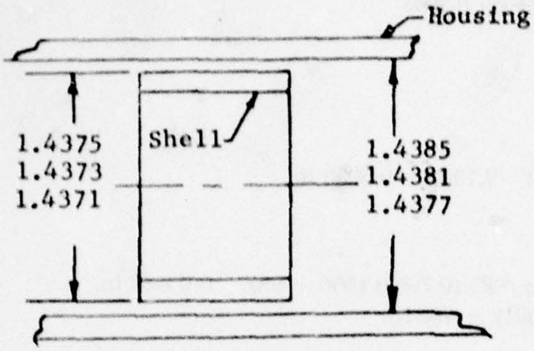
Side plate total length cold = (2) (0.28125) + (2) (0.34075) = 1.244
$\Delta D = (12.98 \times 10^{-6}$ in./in./°F) (1.244)(300 − 68) = 0.003746
Side plate total length hot = 1.244 + 0.0037 = 1.2477

Total Flow Area = [1.2477 − (2) (0.3744)] (0.0041927) = 0.002316

½ total side plate leakage area = 0.002316/2 = 0.0011578

3. Clearance Between Shell and Housing

a.



Cold Clearance

Max	*Nominal*	*Min*
1.4385	1.4381	1.4377
−1.4371	−1.4373	−1.4375
0.0014	0.0008	0.0002

b. Clearance at Running Conditions

Shell α (AMS 4120) at 300°F is 12.98×10^{-6} in./in./°F

$\Delta D = \alpha D \Delta T$

$\Delta D = (12.98 \times 10^{-6}$ in./in./°F) (1.4375) (300 − 68) = 0.004329 = 0.0043

(1.4373) 0.004328

(1.4371) 0.004328

New Shell OD = 1.4375 + 0.0043 = 1.4418

1.4373 1.4416

1.4371 1.4414

Housing α (AMS 4215) at 300°F is 12.7×10^{-6} in./in./°F

$\Delta D = \alpha D \Delta T$

$\Delta D = (12.7 \times 10^{-6}$ in./in./°F) (1.4385) (300 − 68) = 0.0042384 = 0.0042

(1.4381) 0.0042372

(1.4377) 0.0042360

New Housing ID = 1.4385 + 0.0042 = 1.4427

1.4381 1.4423

1.4377 1.4419

Hot Clearance

Max	*Nominal*	*Min*
1.4427	1.4423	1.4419
1.4414	1.4416	1.4418
0.0013	0.0007	0.0001

c. Actual Cold Clearance from Build Measurements

Clearance = 1.4385 − 1.4376 = 0.0009 in.

d. Running Clearance

ΔD shell = (12.98 × 10^{-6} in./in./°F) (1.4376) (300 − 68) = 0.004329
$D housing = (12.9 × 10^{-6} in./in./°F) (1.4385) (300 − 68) = 0.004305

$$DH = \frac{4A}{WP} = \frac{(4)\ (0.000723)}{(2)\ (0.8036) + (2)\ (0.0009)} = 1.7974$$

Running Area = 0.0009 × 0.8036 = 0.000723 in.^{2} Each Side

OD shell 300°F = 1.4376 + 0.0043 = 1.4419
ID housing 300°F = 1.4385 + 0.0043 = 1.4428

Running Clearance = 1.4428 − 1.4419 = 0.0009

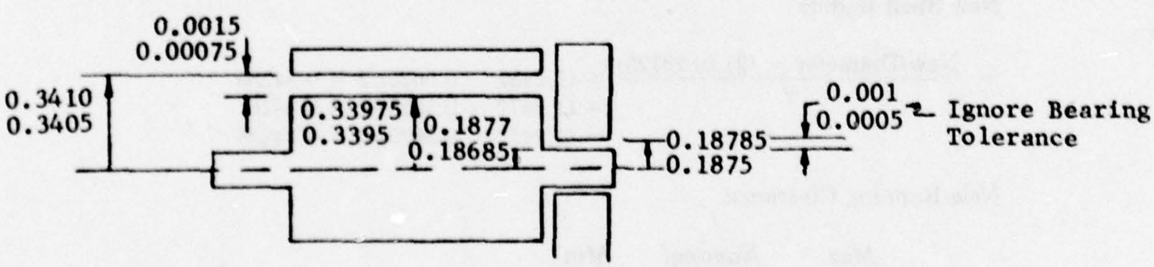
4. Gear Teeth Clearance

a. Notes

(I) Ignore journal tolerance since this could result in a negative clearance. Assume gear shaft is running in center of journal.

(II) Ignore mismatch between end plates and shell since a mismatch that closes down clearance on one side of the pump will open up clearance on other side of pump.

b. Cold Clearances



Cold Clearance on each contracting tooth =

0.3410	−	0.3395	=	0.0015	max
0.34075	−	0.339625	=	0.001125	nom
0.3405	−	0.33975	=	0.00075	min

c. Running (300°F) Clearances

Gear Dimentrical Growth

$$\Delta D = \alpha D \Delta T$$

$$\begin{array}{llll} \Delta D = (6.6 \times 10^{-6} \text{ in./in./°F}) & (0.6795)\,(300 - 68) & = 0.00104045 & \\ & (0.67925) & 0.00104007 & = 0.0010 \\ & (0.6790) & 0.00103968 & \end{array}$$

$$\begin{array}{llll} \text{Running Gear Diameters} = 0.6805 & \text{Radius} = 0.34025 & \approx 0.34025 \\ \phantom{\text{Running Gear Diameters} =} 0.68025 & \phantom{\text{Radius} =} 0.340125 & \approx 0.34013 \\ \phantom{\text{Running Gear Diameters} =} 0.6800 & \phantom{\text{Radius} =} 0.3400 & \approx 0.34000 \end{array}$$

Shell Growth

$$\begin{array}{llll} & & \underline{\textit{New Diameter}} & \\ \text{Diameter} = (2)\,(0.28125) + (2)\,(0.3410) & = 1.2445 & 1.2482 & = 1.2445 + 0.0037 \\ \phantom{\text{Diameter} = (2)\,(0.28125) + (2)\,} (0.34075) & = 1.244 & 1.2477 & = 1.244 + 0.0037 \\ \phantom{\text{Diameter} = (2)\,(0.28125) + (2)\,} (0.3405) & = 1.2435 & 1.2472 & = 1.2435 + 0.0037 \\ & & \uparrow & \end{array}$$

$$\Delta D = \alpha D \Delta T$$

$$\begin{array}{llll} \Delta D = (12.98 \times 10^{-6}) & (1.2445)\,(300 - 68) & = 0.0037476 & \\ & (1.244) & 0.0037461 & = 0.0037 \\ & (1.2435) & 0.0037446 & \end{array}$$

New Shell Radius

$$= \frac{\text{New Diameter} - (2)\,(0.28125)}{2} \quad \begin{array}{l} = (1.2482 - 0.5625)/2 = 0.34285 \\ = (1.2477 - 0.5625)/2 = 0.3426 \\ = (1.2472 - 0.5625)/2 = 0.34235 \end{array}$$

New Running Clearance

Max	*Nominal*	*Min*
0.34285	0.342600	0.34235
−0.34000	−0.340125	−0.34025
0.00285	0.002475	0.00210

Now assume shaft runs on outside of end plate hole under high pressure. This will decrease clearance 0.001 in. making nominal clearance 0.001475 in.

Flow area through teeth is (0.7982) (0.001475) = 0.0011773 in.^{2}

Note we can have three teeth in contact so we have three orifices in series.

$$\frac{1}{A_e^2} = \frac{1}{A_1^2} + \frac{1}{A_2^2} + \frac{1}{A_3^2} = \frac{1}{(0.001173)^2} + \frac{1}{(0.001173)^2} + \frac{1}{(0.001173)^2}$$

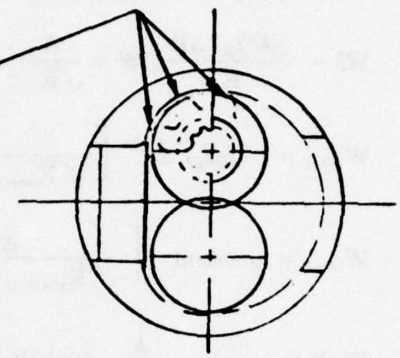
$$\frac{1}{A_e^2} = 2180345.06$$

Effective area through (3) teeth

$$A_e = 6.7723 \times 10^{-4} = 0.00067723 \text{ in.}^2$$

Note that this is for each side of the pump or ½ leakage. For total leakage through the pump

$$A_{teeth} = (2)\ (0.00067723) = 0.00135446$$



From Figure K-2 for the ST-9 pump comparing flow at 150 psid and 0 psid (extrapolated), we have the total leakage.

$$\text{Leakage} = 48.0 \text{ lb/min} - 40.5 \text{ lb/min} = 7.5 \text{ lb/min}$$

$$\tfrac{1}{2} \text{ of leakage} = \frac{7.5}{2} = 3.75$$

$$\Delta P \sim \frac{\rho v^2}{2_{gc}} \sim \frac{W^2}{\rho A^2 2_{gc}}$$

$$W^2 \sim \Delta P\ \rho A^2 2_{gc}$$

$$W \sim \sqrt{\rho}$$

Correcting oil flow from Type II to Type I (7808) at 300°F

$$W_{7807} = W_{II} \sqrt{\frac{\rho_{7808}}{\rho_{II}}} = W_{II} \sqrt{\frac{53.48}{56.38}} = W_{II}\ (0.973942)$$

Zero ΔP flow (no leakage) = (0.973942) (48.0) = 46.749

150 psi ΔP flow = (0.973942) (40.5) = 39.44466

$$\text{Leakage} = 46.75 - 39.44 = 7.31 \text{ lb/min}$$

$$\tfrac{1}{2} \text{ leakage} = \frac{7.31}{2} = 3.66 \text{ lb/min}$$

$$\text{Flow per inch of pump without leakage} = \frac{46.75 \leftarrow \text{flow at } 0\ \Delta P}{0.79942 \leftarrow \text{element length}}$$

= 58.48 lb/min (7808)/inch length

$$W_{\text{leakage total}} = W_{\text{shell to housing}} + W_{\text{gear teeth}} + W_{\text{end plate}}$$

$$\Delta P = K\,\frac{\rho V^2}{2_{gc}} = K\,\frac{W^2}{\rho A^2 2_{gc}}$$

$$W^2 = \frac{\rho A^2 2_{gc}\ \Delta P}{K},\ W = \frac{A}{\sqrt{K}}\ \sqrt{\rho 2_{gc}\ \Delta P}$$

$$W_{\text{total}} = \sqrt{\rho 2_{gc}\ \Delta P}\left[\frac{A}{\sqrt{K_{\text{shell to housing}}}} + \frac{A}{\sqrt{K_{\text{gear teeth}}}} + \frac{A}{\sqrt{K_{\text{end plate}}}}\right]$$

$$W_{\text{total}} = \text{constant}\left[\frac{A}{\sqrt{K_{\text{shell to housing}}}} + \frac{A}{\sqrt{K_{\text{gear teeth}}}} + \frac{A}{\sqrt{K_{\text{end plate}}}}\right]$$

Shell to housing $\frac{A}{\sqrt{K}}$ for half of pump

Approximate length of shell to housing leakage path

$$L = \frac{\pi\ 1.4373}{2} - \frac{0.750}{2} - \frac{0.625}{2} = 2.2577 - 0.375 - 0.3125 = 1.5702$$

assume f = 0.02

$$K = 1.5 + \frac{fL}{D} + K_L = 1.5 + \frac{(0.02)\ (1.5702)}{1.4373} + 0.62 = 1.5 + 0.02 + 0.62 = 2.14$$

$$\frac{A}{\sqrt{K}} = \frac{0.000723}{\sqrt{2.14}} = 0.0004942 \qquad \frac{L}{D_H} = \frac{1.5702}{1.7974}$$

$$= 0.8736$$

$K_L = 0.62$ From Product Engineering "Flow Resistance in Piping and Components," page 15

End plate to gear $\frac{A}{\sqrt{K}}$ for half of pump

$$\text{Treat loss as orifice } K = \frac{1}{C_D^2} = \frac{1}{(0.6)^2} = \frac{1}{0.36} = 2.778$$

$$\frac{A}{\sqrt{K}} = \frac{0.0011578}{\sqrt{2.778}} = 0.0006947$$

Gear teeth leakage $\frac{A}{\sqrt{K}}$ for half of pump

Treat loss as orifice $K = \frac{1}{C_D^2} = \frac{1}{(0.6)^2} = \frac{1}{0.36} = 2.778$

$$\frac{A_e}{\sqrt{K}} = \frac{0.00067773}{\sqrt{2.778}} = 0.0004063$$

$$\text{Constant} = \frac{W(\frac{1}{2}\ \text{leakage})}{\left[\frac{A}{\sqrt{K_{\text{shell to housing}}}} + \frac{A}{\sqrt{K_{\text{gear teeth}}}} + \frac{A}{\sqrt{K_{\text{end plate}}}}\right]} \quad \text{For } \tfrac{1}{2} \text{ of pump}$$

$$\text{Constant} = \frac{3.66}{0.0004942 + 0.0004063 + 0.0006947}$$

Constant = 2294.38

1. ½ *shell to housing leakage* = (2294.38) (0.0004942) = 1.1339

   Total shell to housing leakage = (2) (1.1339) = 2.268

   Total shell to housing leakage per inch of pump = 2.268/0.79942 = 2.837 lb/min/in. pump

2. ½ *gear teeth leakage* = (2294.38) (0.0004063) = 0.9322

   Total gear teeth leakage = (2) (0.9322) = 1.8644

   Total gear teeth leakage per inch of pump = 1.8644/0.79942 = 2.3322 lb/min/in. pump

3. ½ *end plate leakage* = (2294.38) (0.0006947) = 1.5939 lb/min

   Total end plate leakage = (2) (1.5939) = 3.1878 lb/min

   Check, 1.1339 + 0.9322 + 1.5939 = 3.66 lb/min

   Assume the leakage areas of the new pump are the same as the ST-9 test pump.

Pump Size

Required flow = 152.5 lb/min + 15 percent over capacity
= 152.5 + 22.9 = 175.4 lb/min

(no leakage) ↓ shell to housing leak ↓

175.4 = (58.48 lb/min/in. pump) (L) — (2.837 lb/min/in. pump) (L)

gear teeth leak ↓ end plate leakage ↓

— 2.332 lb/min/in. pump (L) — 3.188

175.4 + 3.188 = 53.311 L

$$L = \frac{178.588}{53.311} = 3.3499 \text{ in.} = 3.35 \text{ in.}$$

Scavenge Pump Size

Required flow = 88.6 lb/min

Ignore leakages due to small pump ΔP

Size pump 2X size with no overcapacity

(2) (88.6) = 177.2 lb/min

177.2 lb/min = (58.48 lb/min/inch length) (L)

$$L = \frac{177.2}{58.48} = 3.03 \text{ in.}$$

Derivation of Gear Pump Bearing Journal Loads

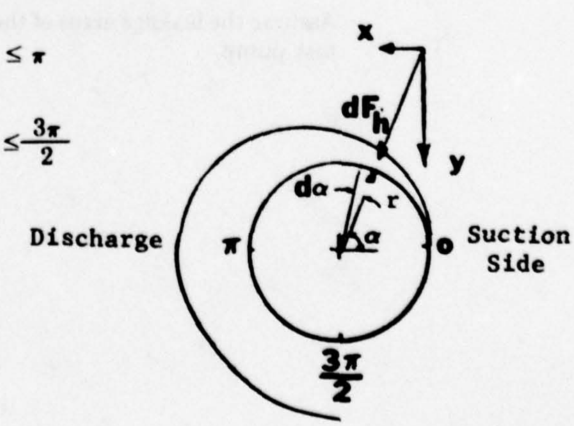
For low pressure applications up to ≈200 psi

1. Assume pressure varies linearly from

$\alpha = 0$ to π and constant from $\alpha = \pi$ to $3\pi/2$

$$P = P\,\text{max}\,\frac{\alpha}{\pi} \qquad 0 \le \alpha \le \pi$$

$$P = P\,\text{max} \qquad \pi \le \alpha \le \frac{3\pi}{2}$$



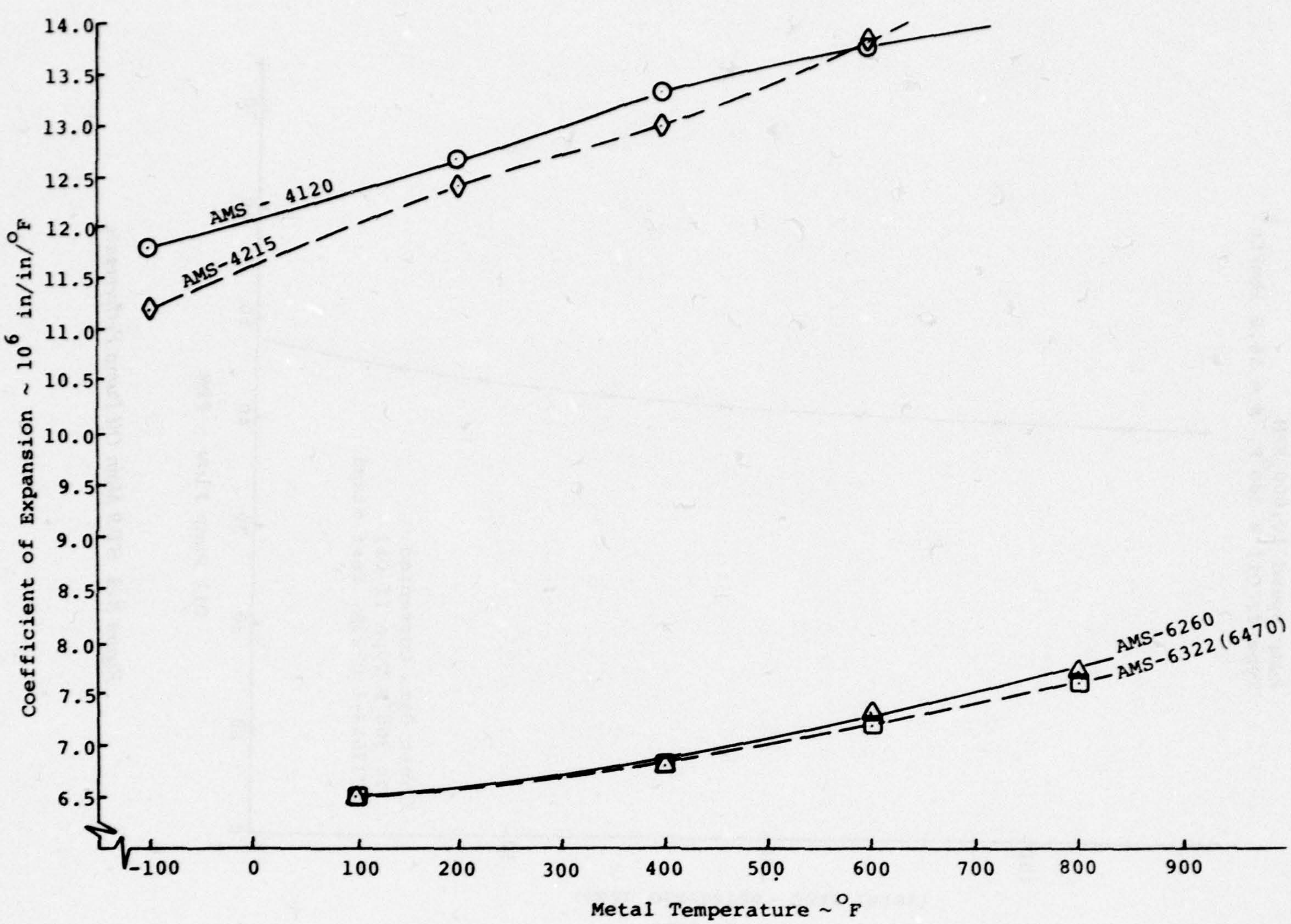


Figure K-1. Coefficient of Thermal Expansion of Pump Materials

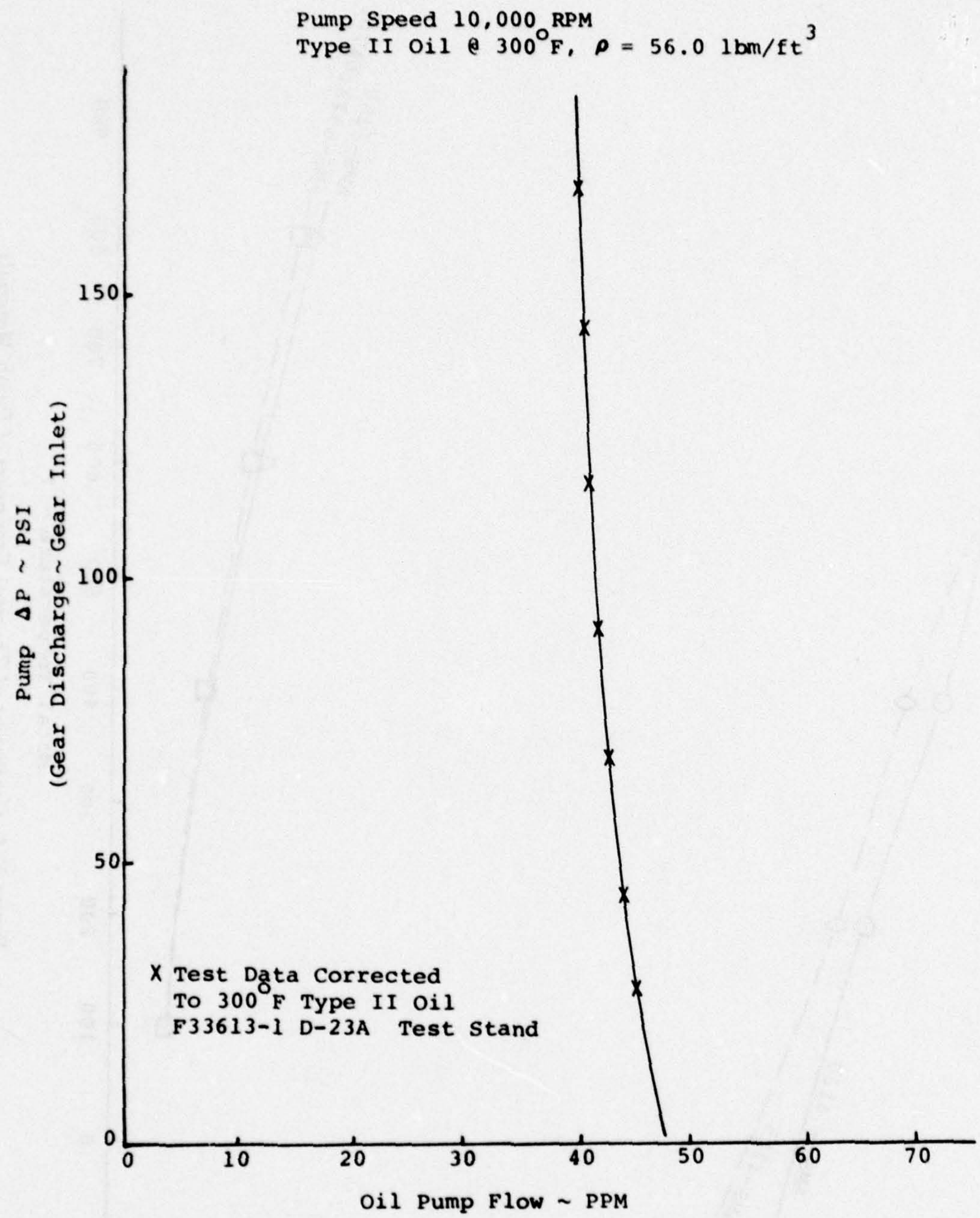


Figure K-2. ST-9 Main Oil Pump Performance

2. **Hydraulic Load**

$0 \leq \alpha \leq \pi$

$$dF_h = PdA = \left(P_{max} \frac{\alpha}{\pi} \right) r\, d\alpha\, \overset{\text{gear width}}{\overset{\downarrow}{w}} = wr \frac{P_{max}}{\pi} \alpha\, d\alpha$$

$\pi \leq \alpha \leq \frac{3\pi}{2}$

$$dF_h = P_{max} (r\, d\alpha\, w) = wr\, P_{max}\, d\alpha$$

3. $dF_{hy} = dF_h \sin \alpha$

$0 < \alpha < \pi$

$$dF_{hy} = wr \frac{P_{max}}{\pi} \alpha \sin \alpha\, d\alpha$$

$\pi < \alpha < \frac{3\pi}{2}$

$$dF_{hy} = wr\, P_{max} \sin \alpha\, d\alpha$$

$$\therefore \quad dF_{hy\ total} = wr \frac{P_{max}}{\pi} \alpha \sin \alpha\, d\alpha + wr\, P_{max} \sin \alpha\, d\alpha$$

$$dF_{hy\ total} = \int dF_{hy\ total} = wr \frac{P_{max}}{\pi} \int_0^{\pi} \alpha \sin \alpha\, d\alpha + wr\, P_{max} \int_{\pi}^{3\pi/2} \sin \alpha\, d\alpha$$

$$\int \alpha \sin \alpha = \sin \alpha - \alpha \cos \alpha$$

$$\therefore \quad F_{hy\ total} = wr \frac{P_{max}}{\pi} (\sin \alpha - \alpha \cos \alpha) \Big|_0^{\pi} - wr\, P_{max} \cos \alpha \Big|_{\pi}^{3\pi/2}$$

$$F_{hy\ total} = wr \frac{P_{max}}{\pi} [(0+\pi) - (0-0)]$$

$$F_{hy\ total} = wr\, P_{max} - wr\, P_{max} = 0$$

The y components therefore cancel each other

4. $dF_{hx} = dF_h \cos\alpha$

$0 < \alpha < \pi$

$$dF_{hx} = wr \frac{P_{max}}{\pi} \cos\alpha \, d\alpha$$

$\pi < \alpha < \frac{3\pi}{2}$

$$dF_{hx} = wr \, P_{max} \cos\alpha \, d\alpha$$

$$dF_{hx\ total} = wr \frac{P_{max}}{\pi} \alpha \cos\alpha \, d\alpha + wr \, P_{max} \cos\alpha \, d\alpha$$

$$F_{hx\ total} = wr \frac{P_{max}}{\pi} \int_0^{\pi} \alpha \cos\alpha \, d\alpha + wr \, P_{max} \int_{\pi}^{3\pi/2} \cos\alpha \, d\alpha$$

$$\int \alpha \cos\alpha \, d\alpha = \cos\alpha + \alpha \sin\alpha$$

$$\therefore \quad F_{hx\ total} = wr \frac{P_{max}}{\pi} (\cos\alpha + \alpha \sin\alpha) \Big|_0^{\pi} + wr \, P_{max} \sin\alpha \Big|_{\pi}^{3\pi/2}$$

$$F_{hx\ total} = wr \frac{P_{max}}{\pi} (-1 + 0 - 1 - 0) + wr \, P_{max} (-1 - 0)$$

$$F_{hx\ total} = wr \, P_{max} \left(-\frac{2}{\pi} - 1\right)$$

$F_{hx\ total} = +1.636 \ wr \, P_{max}$ toward pump inlet

5. **The Gear Forces Are Calculated As Follows:**

Pump HP

$$HP = \frac{144 \ \dot{m} \ \Delta P}{(60)(550) \ \rho\eta}$$

$\dot{m}$ = oil flow, lb/min
ΔP = pressure rise across pump, psi
ρ = oil density, lbm/ft³
η = pump efficiency
N = pump speed, rpm

Pump Torque

$$T = \frac{(33000)(12)(HP)}{2\pi N}$$

½ of torque is transmitted to driven gear and ½ absorbed by driver gear.

Tangential Load

$$F_t = \frac{1}{2}\,\frac{T}{R} \qquad R = \text{gear pitch radius}$$

Separating Load

$$F_s = F_t \tan\theta \qquad \theta = \text{pressure angle}$$

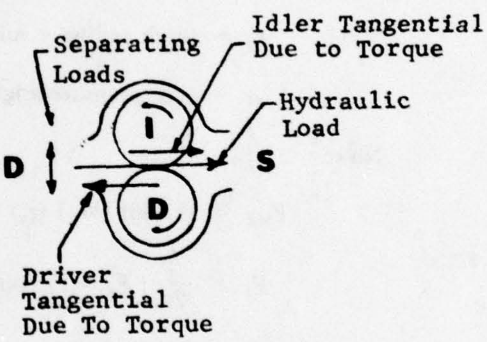
Idler Gear

$$F_{XI} = \text{total load in X direction} = \underset{\text{hydraulic}}{F_{hx}} + \underset{\text{tangential due to torque}}{F_t}$$

$$F_{yI} = \text{total load in y direction} = \underset{\text{separating load}}{F_s}$$

F_I = total load on idler gear

$$F_I = \sqrt{F_{IX}^2 + F_{Iy}^2}$$



Load Diagram

Driver Gear

$$F_{XD} = \text{total load in X direction} = \underset{\text{hydraulic}}{F_{hx}} - \underset{\text{tangential due to torque}}{F_t}$$

$$F_{yD} = \text{total load in y direction} = \underset{\text{separating load}}{F_s}$$

F_D = total load on driver gear

$$F_D = \sqrt{F_{XD}^2 + F_{xy}^2}$$

6. Bearing pressure loads are given by the gear force divided by the projected bearing area.

F100-PW-100 JOURNAL BEARING LOADS

1. F100-PW-100 Pump

$$P = HP = \frac{(144)\ (\Delta P)\ (\dot{m})}{(\rho)\ (\mu)\ (33{,}000)} = 2 \text{ hp}$$

$\dot{m}$ = oil flow in lb/min = 150 lb/min

ρ = density = 59 lb/ft^3

N = pump speed (rpm) = 4072 rpm

W_F = face width (in.) = 1.36

ΔP = pressure across pump = 150 psi max

$$T = \text{torque inch-lb} = \frac{(63{,}000)(2)}{(13{,}900)\ (0.293)} = 30.93 \text{ in.-lb}$$

r_p = gear pitch radius = 0.584 in.

r_o = pitch radius + addendum = 0.750

α = gear pressure angle = 28°

Now:

$$F_{HX} = (1.636)\ (W_F)\ (r_o)\ (\Delta P)$$

$$F_t = \frac{T}{2r_p};\ F_s = F_t\ (\tan\alpha)$$

$$F_{IX} = F_{HX} + F_t$$

$$F_{IY} = F_s$$

$$F_I = \sqrt{F_{Ix}^2 + F_{Iy}^2} = \text{lb load on idler gear}$$

Therefore:

F100-PW-100 journal loading

$$T = 30.93 \text{ in.-lb}$$

$$F_{HX} = 1.636 \times 1.36 \times \frac{1.5}{2} \times 150 = 250.3 \text{ lb}$$

$$F_t = \frac{30.93}{2(0.584)} = 26.48 \text{ lb}$$

$$F_s = 26.48\ (\tan 28°) = 14.08 \text{ lb}$$

$$F_{IX} = 250.3 + 26.48 = 276.78 \text{ lb}$$

$$F_{IY} = F_s = 14.08 \text{ lb}$$

$$\therefore \quad F_I = \sqrt{276.78^2 + 14.08^2} = 277.14 \text{ lb/journal}$$

$$\text{Press load on journal} = \frac{277.14}{(2)(0.455)(0.686)} = 443.9 \text{ psi}$$

2. Scavenge Pump Journal Size

$$HP = \frac{144 \times 15 \times 150}{59 \times 1.00 \times 33{,}000} = 0.166$$

$$\dot{m} = 150 \text{ lb/min}$$

$$\rho = 59 \text{ lb/ft}^3$$

$$N = 10{,}000 \text{ rpm}$$

$$W_F = 3.030$$

$$\Delta P = 15 \text{ psi}$$

$$T = \text{Torque} = \frac{(63{,}000)(0.166)}{10{,}000} = 1.046 \text{ in.-lb}$$

$$r_p = 0.281 \text{ in.}$$

$$r_o = 0.340 \text{ in.}$$

$$\alpha = 28°$$

Now:

$$F_{Hx} = 1.636 \times 3.030 \times 0.340 \times 15 = 25.28 \text{ lb}$$

$$F_t = \frac{1.046}{(2)(0.281)} = 1.86 \text{ lb}$$

$$F_s = 1.86 \tan 28° = 0.989 \text{ lb}$$

$$F_{Ix} = 25.28 + 0.989 = 26.269 \text{ lb}$$

$$F_{Iy} = F_s = 0.989 \text{ lb}$$

$$F_I = \sqrt{(26.269)^2 + (0.989)^2} = 26.28 \text{ lb max gear load}$$

$$\text{Load per journal} = \frac{26.28}{2} = 13.14$$

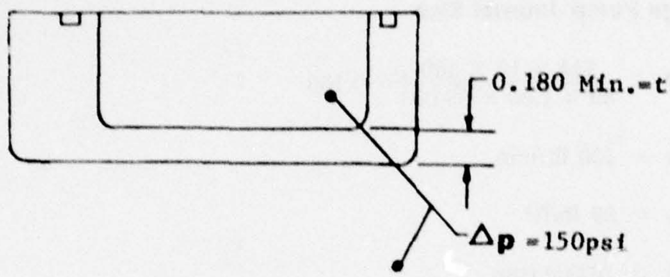
Allowable load from F100-PW-100 pump = 443.9 psi

Allowable pressure load $= \dfrac{\text{Max Journal Load}}{(\text{Dia.})\ (\text{Length})}$

$443.9 = \dfrac{13.14}{(0.373)\ (L)}$ $\qquad$ L = 0.079 in. journal length

From Experience Set Length at 0.250 in.

3. Pump Housing Sample Calculations



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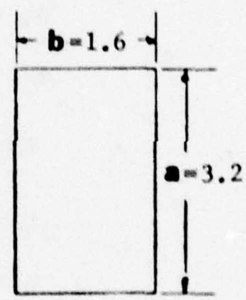


Roark Table X Case 41.
All edges fixed uniform load
over entire surface.

$S_{max} = \beta \dfrac{Wb^2}{t^2}$

$\dfrac{a}{b} = \dfrac{3.2}{1.6} = 2.$; from table Roark page 227, $\beta = 0.497$

W = Load/in.^{2}

a = large dimension of rectangular area under load

b = small dimension of rectangular area under load

t = thickness, in.

$S_{max}\ 0.497 \quad \dfrac{150 \times (1.6)^2}{(0.180)^2} = 5{,}890$ psi bending stress

0.2 percent yield strength of AMS 4117 = 35,000 psi

$\therefore \quad SF = \dfrac{35{,}000}{5{,}890} = 5.94$

4. Line Size for Oil Pump

Inlet Line Velocity (Pressure Pump)

$$V \approx 5 \text{ ft/sec (Experience)}$$

$$W = 150 \text{ lb/min}$$

$$V = 5 \text{ ft/sec}$$

$$\rho = 55 \text{ lb/ft}^3$$

$$W = \rho A V$$

$$A = \frac{W}{\rho V} \quad \frac{\text{lb/min}}{\text{lb/ft}^3 \times \text{ft/sec} \times \frac{60 \text{ sec}}{\text{min}}} = \text{ft}^2$$

$$A = \frac{150}{55 \times 5 \times 60} = 0.00909 \text{ ft}^2$$

$$A = \pi \frac{\rho D^2}{4}$$

$$D^2 = \frac{4 \times 0.00909}{\pi} =$$

$$D = 0.107 \text{ ft} \times 12 = 1.29 \text{ in.}$$

Use 1.250 with 0.035 wall

$$ID = 1.180$$

$$\text{Actual } V = \frac{150 \times 144 \times 4}{55 \times \pi\,(1.180)^2 \times 60} = 5.98 \text{ ft/sec}$$

Pressure Line

$$V \approx 15 \text{ ft/sec Allowable (Experience)}$$

$$W = \rho A V$$

$$A = \frac{150}{55 \times 15 \times 60} = 0.003 \text{ ft}^2$$

$$A = \frac{\pi D^2}{4}$$

$$D^2 = \frac{0.003 \times 4}{\pi} = 0.0038$$

$$D = 0.0618 \times 12 = 0.742 \text{ in.}$$

$$\text{Use 0.750 tubing} \qquad 0.750 - \underset{\uparrow\ 0.035\ \text{wall}}{0.070} = 0.680 \text{ dia}$$

$$\text{Actual V} = \frac{150 \times 144}{55 \times \frac{\pi\ (0.680)^2}{4} \times 60} = 18 \text{ ft/sec}$$

Use same size scavenge inlet and pump inlet. Scavenge discharge from pump

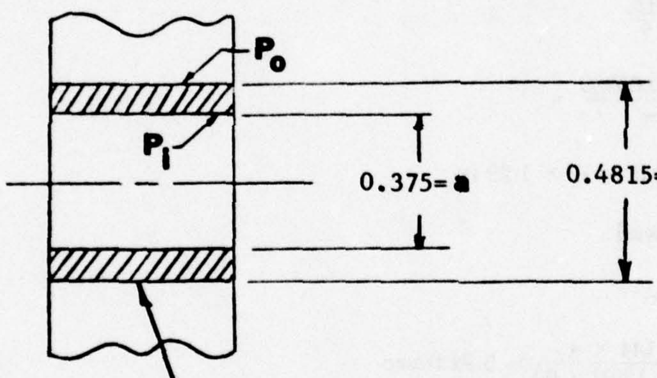
use 0.625 OD tube

ID = 0.555

$$Q = 81.8 \text{ lb/min (2/3 compt) flow}$$

$$V = \frac{81.8 \times 144 \times 4}{55 \times \pi\ (0.555)^2 \times 60} = 14.7 \text{ ft/sec}$$

5. Carbon Bushings Compressive Stress Due to Press Fit



$$\Delta = \frac{P_{ob}}{E}\left[\frac{(b/a)^2 + 1}{(b/a)^2 - 1} - \delta\right]$$

Reference: New Departure Analysis of Stresses and Deflections Copyright 46 by A. B. Jones, page 161

Graphite Carbon Material

$\delta = 0.12$

$E = 3.8 \times 10^6$

$S_{allowable}$ Compressive = 45,000 psi

$\alpha = 2.6 \times 10^{-6}$ in./in./°F

FITS (Carbon to Hsg) ratioed down from F100-PW-100 pump

Max FIT $= 0.0025 + D\alpha\, dt$

$= 0.0025 + (10 \times 10^{-6}$ in./in./°F) (0.4815 in.) (230°F)

Max FIT $= 0.0036$ during operation.

$$\Delta = \frac{P_{ob}}{E}\left[\frac{(b/a)^2 + 1}{(b/a)^2 - 1} - \delta\right]$$

$$0.0036 = \frac{P_o\,(0.4815)}{3.8 \times 10^6}\left[\frac{(0.4815/0.375)^2 + 1}{(0.4815/0.375)^2 - 1} - 0.12\right]$$

$$P_o = \frac{3.8 \times 10^6 \times 0.0036}{(0.4815)(3.96)} = 7175 \text{ psi pressure}$$

$$S_t = \frac{P_i - (b/a)^2 P_o}{(b/a)^2 - 1} + \frac{(P_i - P_o)\, b^2}{4r^2\,[(b/a)^2 - 1]}$$

Reference: New Departure Analysis of Stresses and Deflections, Copyright 46 by A. B. Jones, page 161-163.

$$S_t = \frac{0 - 1.649\,(7175)}{0.649} + \frac{(-7175)\ (0.4815)^2}{4\,(0.24)^2\,(0.648)}$$

$S_t = -18{,}230$ psi $- 11{,}141$

$S_t = -29{,}371$ psi compressive stress

Comp allowable = 45,000 psi
Pure carbon P5Ag
Graphitic carbon

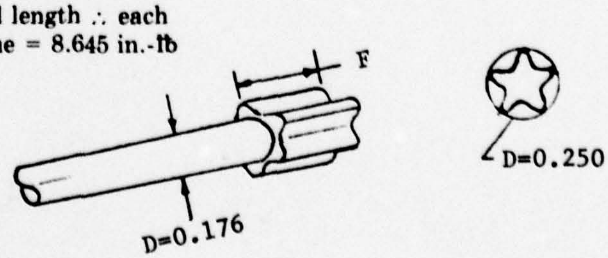
6. Quill Shaft Stresses

$$hp = \frac{144\ W_t\ \Delta P}{33{,}000\ \eta\rho}$$

$hp = 2.746$

$$T = 63{,}000 \times \frac{2.746}{10{,}000} = 17.29 \text{ in.-lb}$$

Make 2 pumps of equal length ∴ each
quill transmits ½ torque = 8.645 in.-lb



$$\text{Spline load} = W = \frac{2T}{D} = \frac{2\ (8.645)}{0.250}$$

$$W = 69.16 \text{ lb}$$

Spline bearing stress

$$S_b = \frac{2T}{D\ N\ F_{h_k}}$$

T = Torque = 8.645 in.-lb
D = Pitch dia = 0.250
N = No. teeth = 11
h_k = Working depth = 0.0128
F = Face width
S_b = 3500 psi allowable (working quill shaft)

$$3500 = \frac{2\ (8.645)}{0.250 \times 11 \times F \times 0.0128}$$

$$F = 0.140$$

Shear Stress in Quill Shaft

Material: H11 Tool Steel

$$S_{shear\ allow} = 0.95\ (200{,}000)\ (0.57) = 108{,}300 \text{ psi}$$

$$S_{shear} = \frac{TL}{J};$$

$$S_{shear} = \frac{16 \times D \times T}{\pi\ (D)^4}$$

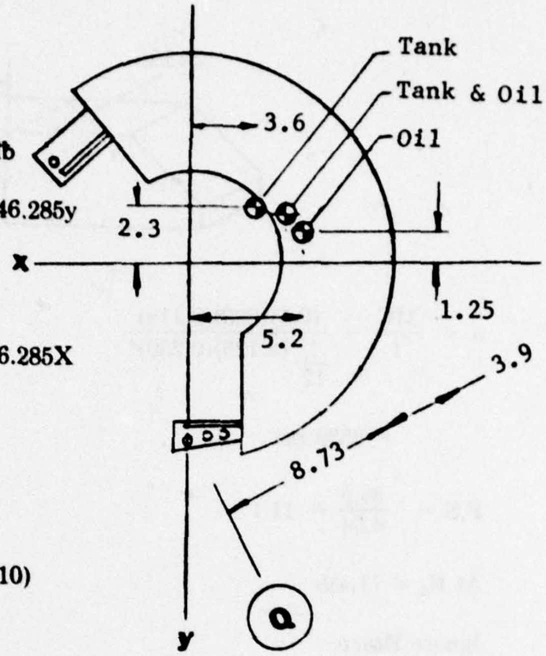
$$S = \frac{16 \times (0.176) \times 8.645}{\pi\ (0.176)^4} = 8076 \text{ psi shear}$$

$$SF = \frac{108{,}300}{8{,}076} = 13.4$$

APPENDIX L

OIL TANK DESIGN

Oil Tank Mount Bracket Stress



Tank $W_T \approx 25$ lb

Oil $W_T = 2.75$ gal $\times$ 7.74 lb/gal = 21.285 lb

$$\Sigma M_x = 0 = 21.285 \times 1.25 + 25 \times 2.3 - 46.285y$$

$$y = \frac{26.6 + 57.5}{46.285} = 1.817$$

$$\Sigma Mg = 0 = 25 \times 3.6 + 21.285 \times 5.2 - 46.285X$$

$$X = \frac{90 + 110.682}{46.285} = 4.336$$

Assume 10g Load

$$\Sigma M_{Q\,Axis} = 0 = 12.63\,R_1 - 46.235(8.73)(10)$$

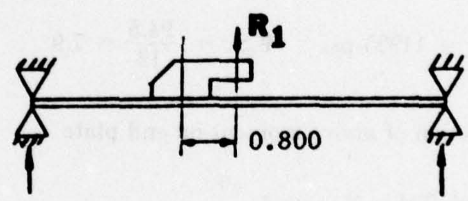
$$\therefore R_1 = 320 \text{ lb}$$

$$\Sigma F \uparrow^{+} = 0 = 320 - 462.85 + 2\,R_2$$

$$R_2 = 71.42 \text{ lb}$$

At R_1 = 320 lb

Reference: Roark Table X, case 5



$r_o = 0.5$
$a = 0.8$

$$S_r \max = \frac{3M}{4\pi t^2 r_o}\left[1 + \left(\frac{M+1}{M}\right) \log 2 \quad \frac{(a - rb)}{Ka}\right]$$

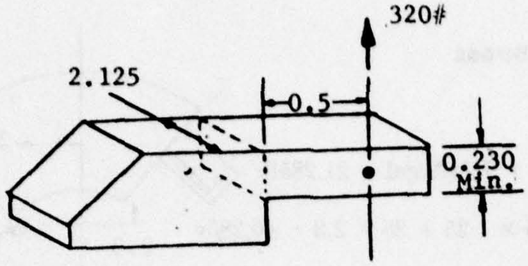
$$K = \frac{0.49a^2}{(r_o + 0.7a)^2} = \frac{(0.49)(0.8)^2}{[0.5 + 0.7(0.8)]^2} = 0.2791$$

$$S_r \max = \frac{3(0.8)(320)(2.28804)}{4\pi(0.062)^2(0.5)}$$

$$= 72{,}754 \text{ psi}$$

S_{allow} = 94,500 psi
AISI 410 at 350°F

$$F.S. = \frac{94.5}{72.7} = 1.299$$



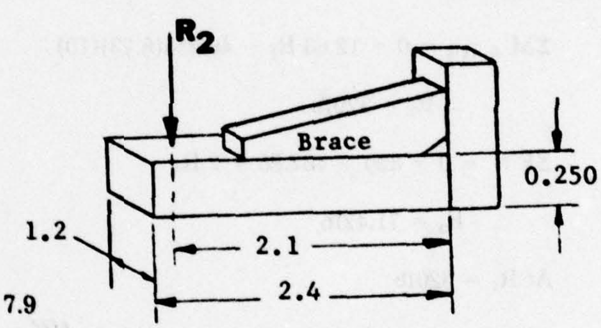
$$\sigma = \frac{MC}{I} = \frac{(0.5)(320)(0.115)}{\frac{1}{12}(2.125)(0.230)^3}$$

$$= 8539 \text{ psi}$$

$$F.S. = \frac{94.5}{8.54} = 11.1$$

At R_2 = 71.4lb

Ignore Brace



$$\sigma = \frac{MC}{I} = \frac{2.1(71.4)(0.125)}{\frac{1}{12}(1.2)(0.25)^3}$$

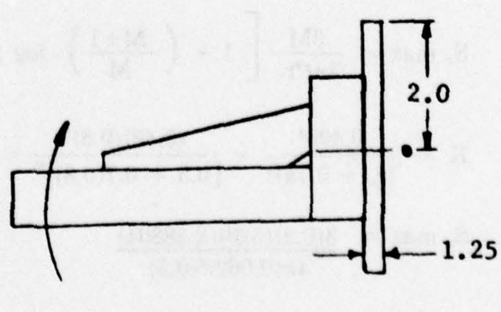
σ = 11995 psi $\quad F.S. = \frac{94.5}{12} = 7.9$

Reaction of above moment on end plate

Roark Table X, case 5

r_o = 0.750

a = 2.0



$$K = \frac{49a^2}{(r_o + 0.7a)^2} = \frac{(0.49)(2)^2}{[(0.75) + 0.7(2)]^2}$$

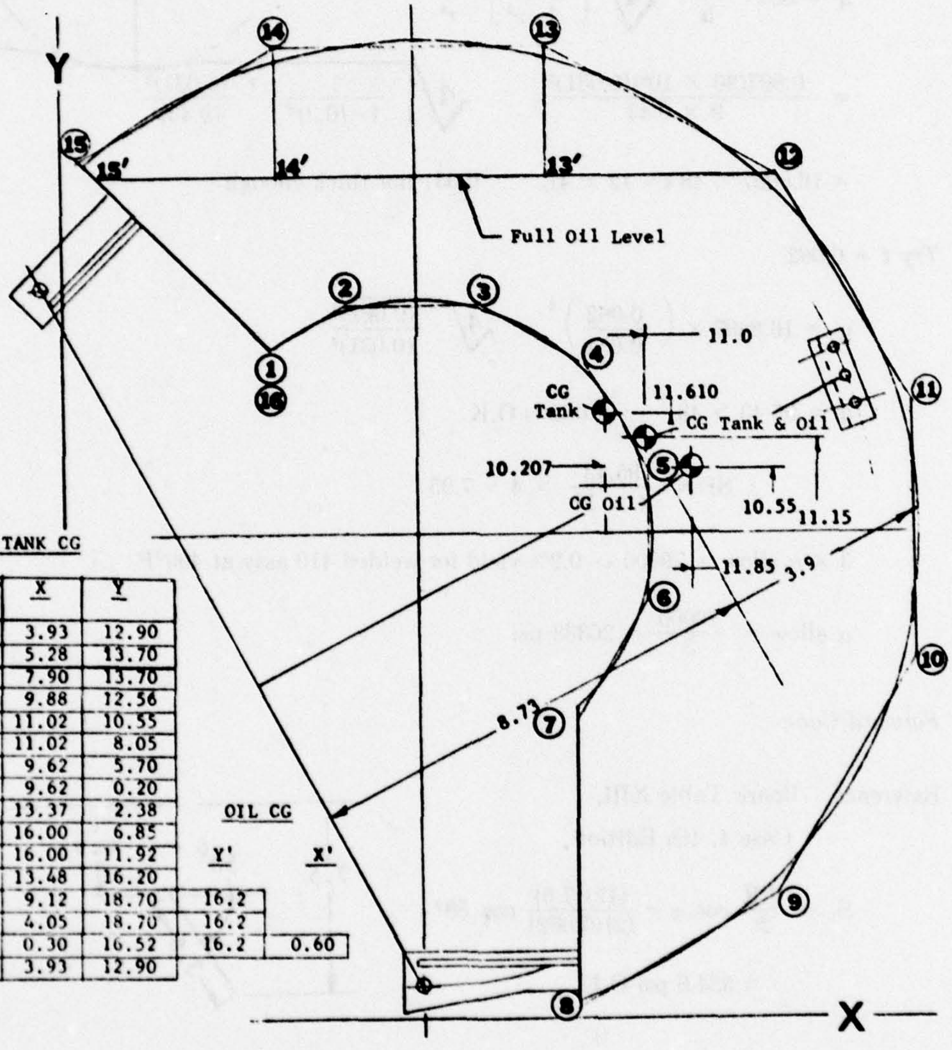
$$= 0.4240$$

$$S_{\sigma\ max} = \frac{3M}{4\pi t^2 r_o}\left[1 + \frac{(M+1)}{M}\ \log \frac{(2)(a - r_o)}{Ka}\right]$$

$$= \frac{3(71.4)(2.1)}{4\pi(0.125)^2(0.75)}\left[1 + \frac{4.3}{3.3}\ \log \frac{(2)(2 - 0.75)}{0.424(2)}\right]$$

$$= 7357.7 \text{ psi}$$

$$\text{F.S.} = \frac{94.5}{7.4} = 12.84$$



OIL TANK CG

POINT	X	Y
1	3.93	12.90
2	5.28	13.70
3	7.90	13.70
4	9.88	12.56
5	11.02	10.55
6	11.02	8.05
7	9.62	5.70
8	9.62	0.20
9	13.37	2.38
10	16.00	6.85
11	16.00	11.92
12	13.48	16.20
13	9.12	18.70
14	4.05	18.70
15	0.30	16.52
16	3.93	12.90

OIL CG

POINT	Y'	X'
13	16.2	
14	16.2	
15	16.2	0.60

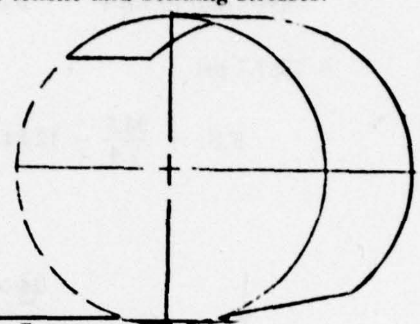
Figure L-1. Oil Tank Center of Gravity Calculations

Buckling Collapsing of Tank Due to External Pressure During Component Tests

The various parts of the oil tank can be considered complete rings.

A minimum safety factor of 4 on buckling and 3 on tensile and bending stresses.

Outer Wall



Reference: Roark Table 35,
Case 196, 5th Edition

$$q' = 807 \ \frac{E_t^2}{lr} \ \sqrt[4]{\left[\frac{1}{1-\nu^2}\right]^3 \frac{t^2}{r^2}}$$

$$= \frac{0.807(30 \times 10^6)(0.031)^2}{9 \times 9.43} \ \sqrt[4]{\left[\frac{1}{1-(0.3)^2}\right]^3 \frac{(0.031)^2}{(9.43)^2}}$$

$= 16.8697 < 48 \ (= 12 \times 4) \quad \therefore$ 0.031 not thick enough

Try t = 0.062

$$q' = 16.8697 \times \left(\frac{0.062}{0.031}\right)^2 \times \sqrt[4]{\frac{(0.062)^2}{(0.031)^2}}$$

$q' = 95.43 > 48 \quad \therefore$ 0.062 is O.K.

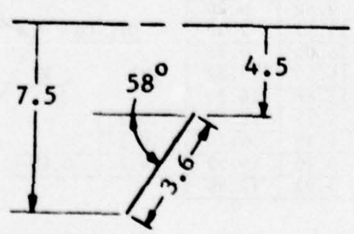
$$\therefore SF = \frac{95.43}{4 \times 12} \times 4 = 7.95$$

$3 \times \sigma$ allow = 79000 ← 0.2% yield for welded 410 assy at 400°F

$$\sigma \text{ allow} = \frac{79000}{3} = 26333 \text{ psi}$$

Forward Cone

Reference: Roark Table XIII,
Case 4, 4th Edition



$$S_1 = \frac{PR}{2t} \cos\alpha = \frac{(12)(7.5)}{(2)(0.062)} \cos 58°$$

= 384.6 psi O.K.

For Axial End Support

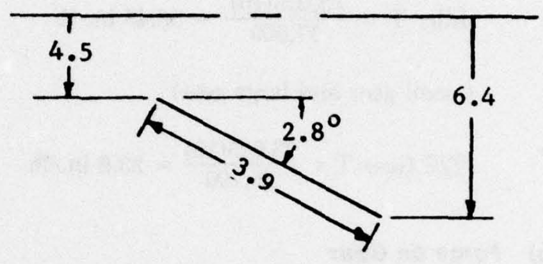
$$S_2 = P\left[\sqrt[4]{12(1-\nu^2)}\ \sqrt{\frac{R^2 \sin\alpha}{2t^2 \cos\alpha} + \frac{(1-\nu/2)R}{t \cos\alpha}}\right] = 0.20040 \text{ psi}$$

For Tangential End Support

$$S_2 = \frac{PR}{t\cos\alpha} = \frac{12(7.5)}{(0.062)(\cos 58^\circ)} = 2739 \text{ psi} \qquad \text{O.K. for 410 SST}$$

Reference: Roark Table XIII
Case 3, 4th Edition

Rear Cone



$$S_1 = \frac{12(6.4)}{2(0.062)} \cos 58^\circ$$

$$= 328 \text{ psi}$$

For Axial End Support

$$S_2 = 12\left[\sqrt[4]{12(0.7)}\ \sqrt{\frac{6.4^2 \sin {}^228^\circ}{2(0.062)^2 \cos 28^\circ} + \frac{(1\ -0.15)(6.4)}{0.062(\cos 28^\circ)}}\right]$$

$$= 12\ (-630 + 99.373) = 12\ (-531.4)$$

$$= -6376.9 \text{ psi}$$

For Tangential End Support

$$S_2 = \frac{PR}{t\cos\alpha} = \frac{12 \times 6.4}{0.062 \cos 28} = 1402 \text{ psi}$$

APPENDIX M
HIGH-SPEED DRIVE TRAIN ANALYSIS

JT9D Gears Used in Compartmental Lube System (Check Gears for Rig Condition)

For Slowest Gear (Pump)

(a) $$T = 63{,}025 \; \frac{\overset{\text{Actually 1.47 hp}}{\downarrow}\,(10 \text{ hp})}{(10{,}000 \text{ rpm})} = 63.025 \text{ in.-lb}$$

$$\text{Idler } T = \frac{63{,}025(10)}{17{,}300} = 36.43 \text{ in.-lb}$$

(small gear and large gear)

$$\text{T/S Gear } T = \frac{63{,}025(10)}{26{,}700} = 23.6 \text{ in.-lb}$$

(b) Force on Gear

$$W = \frac{2T}{D}$$

Pump Gear

$$W = \frac{2(63)}{3.918} = 32.159 \text{ lb}$$

Idler

$$W = \frac{2(36.4)}{2.3} = 31.652 \text{ lb}$$

T/S Gear

$$W = \frac{2(23.6)}{2.9166} = 16.183 \text{ lb}$$

(c) Pitch Line Velocity ('V) = 0.262 (D) (N) ft/min

Pump and Idler (small)

$$V = 0.262\ (3.918)(10{,}000) = 10{,}265 \text{ ft/min}$$

Idler (large) and T/S H Gear

$$V = 0.262(4.50)(17{,}000) = 20{,}043 \text{ ft/min}$$

(d) Minimum Effective Face Width

$$F = \frac{21 \times 10^{6}(W)(mg + 1)}{\sin 2\theta\, D\, (S_c)^2\, mg}$$

For Pump — Small Idler

$$F = \frac{21 \times 10^{6}(32.159)(2.704)}{\sin (45)(2.3)(135{,}000)^2(1.704)} = 0.036; \frac{W}{F} = 889.4$$

For Idler — Towershaft

$$F = \frac{21 \times 10^{6}(35.43)(2.543)}{\sin 45(4.5)(136{,}500)^2(1.543)} = 0.021; \frac{W}{F} = 753$$

Actual F = ***0.170 min***

(e) $$W_d = \frac{0.05V\,[F\,(C) + W]}{0.05V + [F\,(C) + W]^{1/2}} + W$$

For Pump Gear

$$W_d = \frac{0.05(10{,}265)\,[0.021(890) + 32.16]}{0.05(10{,}265) + [(0.021)(890) + 32.16]^{1/2}} + 32.16$$

$$= 50.15 + 32.159$$

$$= 82.3 \text{ lb}$$

(f) $$\frac{W_d}{F} = \frac{82.3}{0.021} = 3919.6$$

At $\frac{W_d}{F} = 3920$, $P_d = 8.3$ **D.P. Actually is 11.7391, so is OK.**

At $R = 0.031$, $K = 1.3$

(g) For Pump Gear X = 0.075 (5905 printout)

(h) $W_e = \underline{0.667(63{,}000)(0.170)(0.075)}$

$$= 412.2 \geq W_d = 82.3$$

(i) N/A

(j) $W_s = \dfrac{0.667 \times 130{,}000(0.170)(0.075)}{1.3} = 850{,}425$

$$850.425 \geq 1.5\ W_d = 123.45$$

(k) No T_s (=0)

Wave Washers (see Figure M-1)

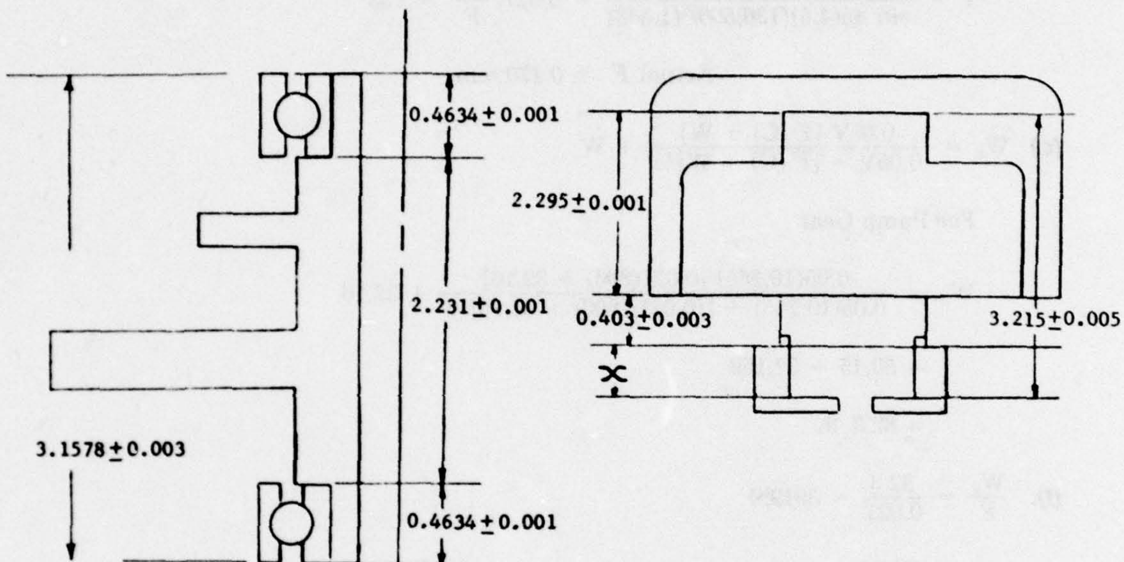


Figure M-1. Idler Shaft Wavewasher Gap

Idler Shaft

50 + 75 lb
OD = 1.65 max
ID = 1.25 min

Use 2 — Associated Springs
P/N W1621-019

Bevel Gear

75 + 100 lb
OD = 2.830 max
ID = 2.2 min

Associated Spring
P/N W2816-030

$65 \text{ lb}/0.1 \text{ in.} = 85 \text{ lb}/\Delta$

$$\Delta = \frac{85}{65} \times 0.1 = 0.13077 \text{ Deflection}$$

$$0.197 - 0.13077 = 0.0662 \text{ Installed Height}$$

Need 0.0715 gap for spring pre-load

Gap = 0.0715

	2.295
3.1578	0.403
+ Gap	+ X
3.229	3.229

$$X = 0.531$$

Calculation of Wavewasher Deflection

Reference: Mechanical Springs
The William D. Gibson Co.
p 93

$$l/f = \frac{E\, b\, t^3 N^4}{P\, 1.94\, D^3}$$

$$b = \frac{1.621 - 1.261}{2} = 0.18$$

$$D = \frac{1.621 + 1.261}{2} = 1.441$$

$P = 32$

$t = 0.0185$

$N = 3$

$E = 30 \times 10^6$

$l/f = 16.963$

$f = 0.05895$

Installed Height = 0.112 − 0.05895 = 0.053

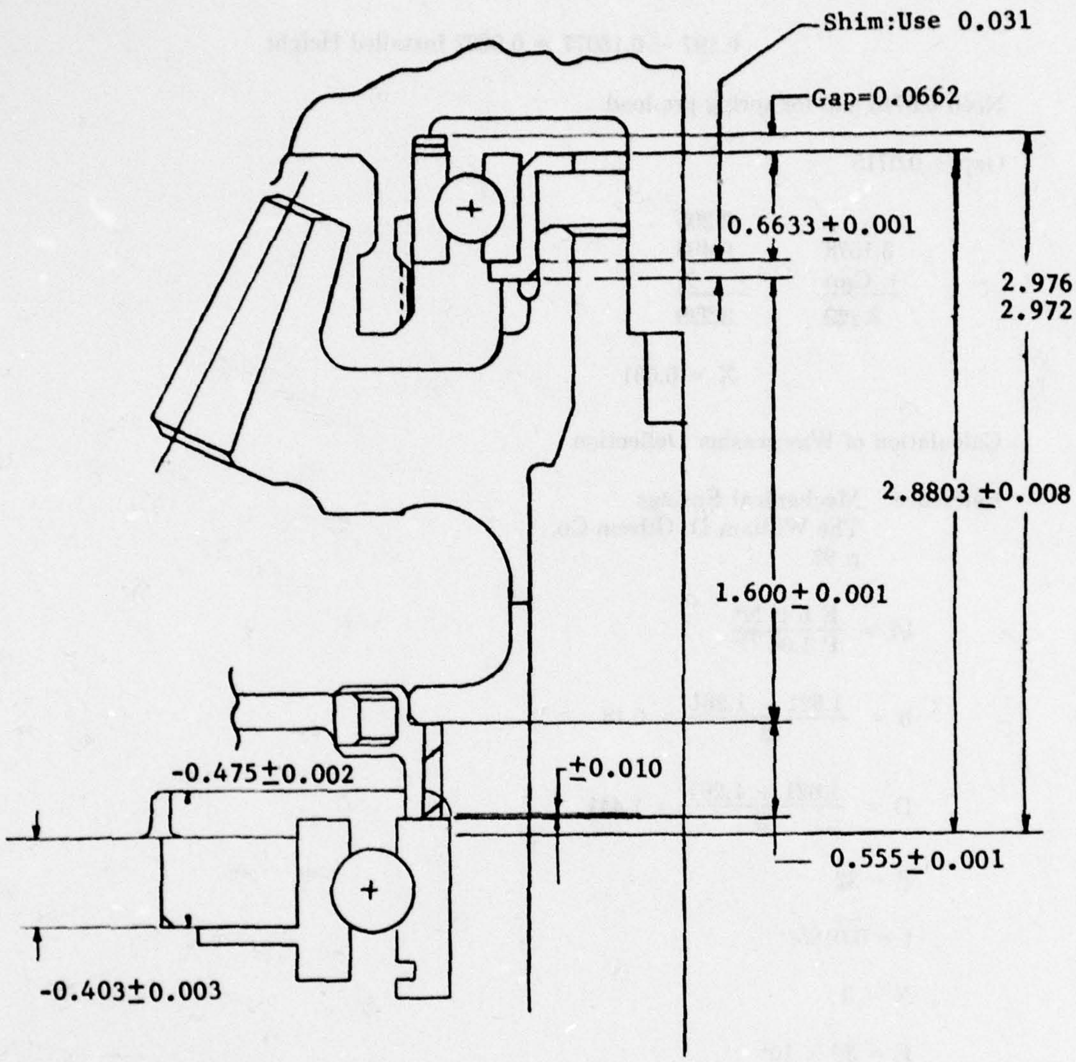


Figure M-2. Towershaft Wavewasher Gap

Installed Height for 2 wavewashers in series

$$= 0.053 + \text{washer thickness}$$
$$= 0.053 + 0.0185 = 0.0715$$

This provides a 64 lb pre-load.

PIPE SUPPORT FLANGE STRUCTURAL ANALYSIS

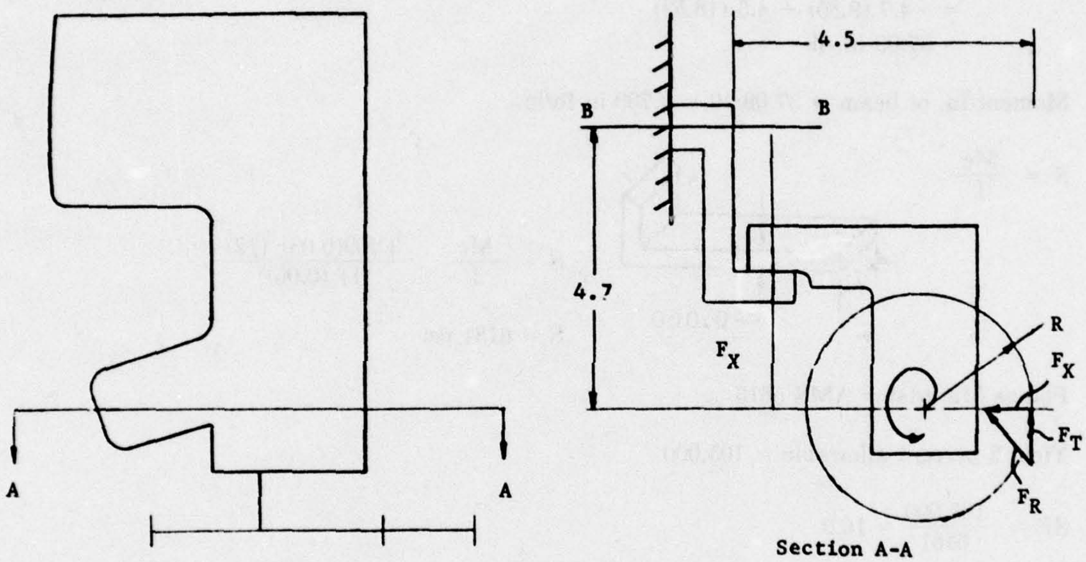


Figure M-3. Pipe Support Flange Structural Analysis

$$T = 63{,}000 \ \frac{hp}{rpm}$$

T = Torque in.-lb

hp = total horsepower = 5 (high for safety)

rpm = pump speed = 10,000 rpm

$$T = 63{,}000 \ \frac{5}{10{,}000} = 31.5 \text{ in.-lb}$$

$$F_T = \text{Tangential load} = \frac{T}{R} = \frac{31.5}{1.7} = 18.53 \text{ lb}$$

$R = 1.7$ in.

$F_x = \tan \phi \, (18.53)$

ϕ = pressure angle = 28°

$F_x = \text{Tan } 28° \, (18.53) = 9.85$ lb

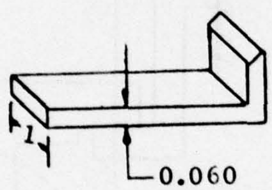
$F_r \quad \sqrt{(9.85)^2 + (18.53)^2} = 20.98$ lb

ΣMo about plane B-B

$$\begin{aligned} \Sigma Mo &= -4.7\,(F_x) + 4.5\,(F_T) \\ &= -4.7\,(9.85) + 4.5\,(18.53) \\ &= 37.09 \text{ in.-lb} \end{aligned}$$

Moment/in. of beam = 37.09/10 = 3.709 in lb/in.

$$S = \frac{Mc}{I}$$



$$S = \frac{Mc}{I} \qquad \frac{3.709(0.03)\ (12)}{(1)\ (0.06)^2}$$

$S = 6181$ psi

Flange Material = AMS 5616

Yield 2 percent allowable = 105,000

$$SF = \frac{105{,}000}{6181} = 16.9$$

APPENDIX N
OIL JET SIZING

$$\Delta P = K \frac{\rho v^2}{2_{gc}} = K \frac{W^2}{\rho A^2 2_{gc}}$$

$$\Delta = 45 \text{ psi}$$

$$W = 1 \text{ lbm/min} = \frac{1}{60} \text{ lbm/sec} = \text{flow per jet}$$

$$\rho = 57.9 \text{ lbm/ft}^3$$

$$gc = 32.2 \frac{\text{ft lbm}}{\text{lb}_f \text{ sec}^2}$$

$$K = 1.5 + f \frac{L}{D} = 1.5 + 0.06 = 1.56$$

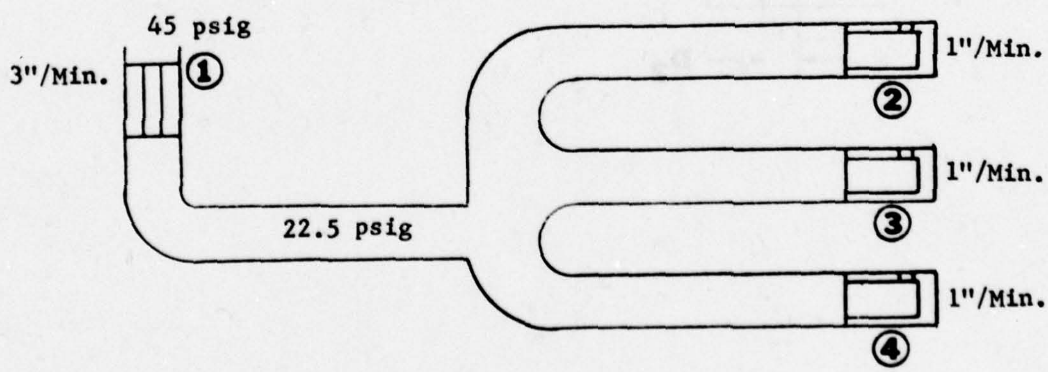
$$A^2 = \frac{KW^2}{\rho \Delta P 2g_c} = \frac{1.56\ (1/60)^2}{(57.9)\ (45)\ (2 \times 32.2)}$$

$$= \frac{1.56\ (144)}{60^2\ (57.9)\ (45)\ (64.4)}$$

$$A = 0.0006098 \text{ in}^2 = \frac{\pi D^2}{4}$$

$$D = 0.0278 \text{ (too small)}$$

Must use upstream jet to reduce ΔP over individual jets



Jet (1) $W_1 = 3 \text{ lb/min} = \frac{1}{20} \text{ lb/sec}; \ W_2 = \frac{1}{60} \text{ lb/sec}$

$\Delta P = 22.5 \text{ psi}$

$\rho = 57.9 \text{ lbm/ft}^3$

$gc = 32.2 \ \frac{\text{ft} \cdot \text{lbm}}{\text{lb}_f \cdot \text{sec}^2}$

$K_1 = 1.5 + 0 + 0.038\,(4) = 1.652$

$K_2 = 1.5 + 0.55\,(.038)\,(1) = 2.088$

Jet (1) $A_1^2 = \frac{KW^2}{\rho \Delta P 2gc} = \frac{1.652\,(1/20)^2}{57.9\,(22.5)\,(2 \times 32.2)\,(1/144)}$

$= 0.0000070886$

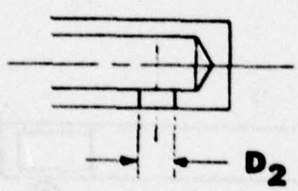
$A_1 = \frac{\pi D^2}{4} = 0.0026624$

$D_1 = \sqrt{0.00338995} = 0.05822 \text{ in.}$

Jet (2) $A_2^2 = A_1^2 \ \frac{(1/60)}{1/20} \left(\frac{2.088}{1.562} \right) = A_1^2\,(1/3)\,(1.3362) = 0.0000031584$

$A = 0.0017772 = \frac{\pi D_2^2}{4}$

$D_2 = \sqrt{0.0022628112} = 0.047569 = D_2 = D_3 = D_4$



APPENDIX O
DATA LOG FOR COMPARTMENTAL LUBRICATION SYSTEM 50-HR ENDURANCE TEST

This appendix contains all of the data recorded during the 50-hr endurance test of the Compartmental Lubrication System Rig.

Pratt & Whitney Aircraft U.A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet 1/4 A
Date 1/23/78
Engineer [illegible]
Operators [illegible]

D-4 Stand [redacted]/Rig No. 34024 Build 10 Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time	Mission Point No.	P21 LAB Cooling	P10 Forward dome	P11 REAR Dome	P3 2,3 OIL	P4 1,4,5 OIL	P2 OIL Pump	P14 Bore Exit	P19 6/8 OIL IN	P20 6/8 OIL out	P5 2,3 Breather	P9 Bore Chamb	P7 Forward Chamber	P8 Rear Chamb	P15 Breather orifice	P16 2,3 Scavenge	P17 Forward Dome orifice	P18 Rear Dome orifice	P1 OIL TANK	P6 1,4,5 AIR	ΔP2	ΔP16
		PSIG	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA		
		G-2	G11	G12	G13	G14	G15	G16	G17	G18	G19	G20	G21	G22	G37	G38	G39	G40	G41	G42		
0815	2	0	0	0	14.7	14.7	14.7	14.7	0	0	14.7	14.7	14.7	14.7	14.7	14.7	14.7	15.0	15.0	14.7	–	–
0900		Start Rotation																				
[illegible]		ON En[illegible]																				
[illegible]		21	0	10	[illegible]	[illegible]	115	15	13	0	14.2	18.5	15	18	[illegible]	[illegible]	17.0	20	13.5	[illegible]	.50	0
[illegible]		OFF Endurance																				
[illegible]		46	34	19	75	56	170	19	14	0	14	39	[illegible]	[illegible]	6.7	24.0	53.5	52.0	13.0	152	2.00	
[illegible]	2	ON E[illegible]																				
1038		[illegible]	22	25	76	57	190	18	14	0	14	35	24	28	7.6	25	[illegible]	[illegible]	13	158	5.00	
1040		OFF E[illegible]																				
1045		Stop		CHIP DET LIGHT																		
1445		Start Rotation																				
1500	1	On Endurance																				
1505		[illegible]	[illegible]	1.0	44	29	113	16	14	0	14.2	18	15.5	16	13.5	17	18	21	13.5	[illegible]	.1	0
1530		25	2.0	5.0	43	28.5	108.5	14	14	0	14.1	18	15	17	13.7	14.8	17.5	23.5	13.5	67.5	.1	0
1600		25	2.0	5.0	43	28.5	112	15	14	0	14.1	18	15.5	17.5	13.7	15.5	18.	23.5	13.5	67	.1	0
1630		25	2.0	5.0	43	28.5	112	15	14	0	14.0	18	15.5	17.5	13.7	15.6	18	23.5	13.5	67	.1	0
1700		25	2.0	5.0	43	28	107	15	14	0	14.0	18	15.5	17.5	13.7	15.7	18	23.5	13.5	66.5	.1	[illegible]
1730		25	2.0	5.0	43	28	108	15	14	0	14.0	18	15.5	17.5	13.7	15.6	18	23.5	13.5	66	.1	[illegible]
1800		25	2.0	6.0	43	28.5	107.5	15	14	0	14	18	15.5	17.5	13.8	15.6	18	24	13.5	67	.1	0
1830		25.5	2.0	6.0	43	28.5	109	15	14	0	14	18.5	15.5	17.5	13.7	15.7	18	24	13.5	67	.1	0
1900		25.5	2.0	6.0	43	28.5	107	15	14	0	14	18.0	15.5	17.5	13.8	15.7	18	24	13.5	66	.1	0
1930		25.5	2.0	6.0	43	28.5	108	15	14	0	14	18	15.5	17.5	13.7	15.7	18	24	13.5	67	.1	0
2000	1	25.5	2.0	6.0	43	28.5	108	15	14	0	14	18.5	15.5	17.5	13.7	15.7	18	24	13.5	68	.1	0

Remarks 0900 → 1045 OIL LOSS = 1½ gal

1045 − 0900 = 1:45 RT   TOT ENDURANCE = 7:15 [illegible]   2000 − 1445 = 5:15 RT   2000 − 1500 = 5:00 END [illegible]

Page No.   Report No.

Pratt & Whitney Aircraft U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 2/4A

Date 1/25/78

Engineer Gamble

Operators Canby/Russell

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs. HIMSELF POINT DRY	"U" Tubes Dp 6 (INCHES HG) U-63	Dp 15 U-64	Dp 17 U-65	Dp 18 U-66		Flow Meters F1 OIL Supply PPM	F2 1,4,5 OIL PPM	F3 2,3 OIL PPM	F4 G/B PPM	Pump Speed RPM	Rig Speed RPM	Vibrations (MILS) RIG NO. 3 BRG 350° V-1	RIG NO. 3 BRG 260° V-2	RIG NO. 2 BRG VERT V-3	RIG FRONT VERT V-4	RIG FRONT HORZ V-5	RIG REAR VERT V-6	RIG REAR HORZ V-7	G/B VERT V-8	G/B HOR. V-9
0815	2	0	0	0	0		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0900		Start Rotation																			
0935		ON Endurance																			
0946		9.5	.3	.5	6.1		121.8	56.9	64.6	11.0	6758	9260	.12	.1	.12	.12	.15	.1	.12	.25	.31
0950		OFF Endurance																			
1020		59.5	2.3	1.8	17.8		147.7	59.0	87.6	11.5	9078	12430	.21	.3	.17	.15	.2	.15	.02	.36	.27
1035	2	ON Endurance																			
1038		66	2.3	2.6	18		143.3	54.3	87.8	11.6	9074	12240	.15	.1	.13	.13	.13	.15	.16	.31	.17
1040		OFF ENDURANCE																			
1045		Stop																			
1445		Start Rotation																			
1500		ON Endurance																			
1505		8.9	.4	.8	3.7		120.0	56.6	63.4	11.5	6706	9180	.14	.17	.11	.18	.22	.01	.17	.2	.22
1530		9.3	.2	.6	4.5		116.5	53.6	62.7	11.5	6636	9090	.14	.18	.1	.15	.22	.05	.20	.25	.26
1600		9.2	.2	.6	4.5		112.5	54.8	62.5	11.5	6611	9020	.14	.17	.1	.15	.22	.05	.20	.24	.27
1630		9.3	.3	.6	4.5		112.2	54.6	62.7	11.5	6614	9060	.14	.17	.1	.15	.21	.05	.20	.23	.26
1700		9.2	.3	.7	4.5		112.5	54.9	62.4	11.5	6647	9090	.13	.17	.09	.15	.21	.05	.19	.23	.27
1730		8.8	.3	.6	4.5		118.7	55.8	62.4	11.5	6612	9060	.12	.16	.09	.15	.20	.05	.19	.23	.26
1800		9.4	.3	.6	4.5		117.1	54.3	62.6	11.4	6624	9070	.14	.17	.09	.15	.22	.06	.20	.23	.26
1830		9.3	.3	.6	4.5		116.2	53.6	62.5	11.5	6628	9080	.14	.17	.09	.15	.22	.06	.20	.22	.26
1900		8.8	.3	.6	4.5		116.5	53.8	62.2	11.5	6653	9110	.14	.16	.09	.15	.22	.07	.20	.22	.26
1930		9.4	.3	.6	4.5		116.3	53.3	62.6	11.4	6652	9120	.14	.16	.1	.15	.22	.07	.20	.21	.25
2000		9.4	.3	.6	4.7		115.9	53.4	62.4	11.5	6628	9070	.14	.16	.1	.15	.22	.07	.20	.21	.25

Page No.

Remarks

PWA FRDC 97 REV 3 71

FD 489

Pratt & Whitney Aircraft

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LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. A'
Date 1/25/78
Engineer Bill Gamble
Operators Conley/Renner

D-4 Stand   Engine/Rig No. F-34024   Build 10   Project

Type of Test 50 HR ENDURANCE RUN

"A" Switch °F

Time A.M. P.M.	Test Hrs / Mission Point No.	T_1 Oil Tank	T_2 Oil Pump Dis.	T_3 2/3 Comp Sup	T_4 1-4-5 Oil Sup.	T_{51} 2/3 Comp Air	T_{52} 2/5 Comp Air	T_6 1-4-5 Air Ori.	T_{71} Forw Cham Temp	T_{72} Forw Cham Temp	T_{81} Rear Cham Temp	T_{82} Rear Cham Temp	T_9 Bore Temp	T_{10} Forw Dome Temp	T_{12} Forw Dome Temp	T_{11} Rear Dome	T_{13} Rear Dome	T_{15} Brea. Air Orif.	T_{16} 2/3 Scav. Rim Disc	T_{14} Bore Air Out
		A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9	A-10	A-11	A-12	A-13	A-14	A-15	A-16	A-17	A-18	A-19
0815	2	70	69	69	69	71	71	69	70	70	70	70	69	70	70	70	70	69	70	[illegible]
0900		START ROTATION																		
0935		ON ENDURANCE																		
0940		214	214	209	208	218	219	76	150	150	192	193	221	92	93	122	122	149	230	[illegible]
[illegible]		OFF ENDURANCE																		
1020		231	231	209	207	243	245	84	342	347	437	438	360	366	366	406	408	210	255	316
1035	2	ON ENDURANCE																		
1038		285	/	264	259	293	296	85	405	411	[illegible]	[illegible]	446	414	[illegible]	[illegible]	[illegible]	244	312	323
1040		OFF ENDURANCE																		
1045		STOP																		
1405		START.																		
1500	1	ON ENDURANCE																		
1505		206	206	201	200	212	212	83	147	147	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	219	143
1530		234	234	231	229	239	239	83	161	161	218	218	240	102	101	143	143	170	243	151
1600		212	212	200	199	217	218	83	159	158	209	209	263	103	102	138	139	170	221	135
1630		215	214	208	208	220	221	82	155	155	205	205	203	101	100	134	135	161	228	145
1700		242	241	233	231	245	245	81	166	166	214	214	203	101	101	135	136	171	247	141
1730		236	236	225	224	240	239	79	163	162	210	210	191	100	100	133	133	172	238	134
1800		237	237	230	229	242	241	79	162	161	209	209	191	97	98	132	133	171	196	135
1830		225	225	220	219	230	232	80	157	157	205	205	221	99	95	132	133	170	238	149
1900		234	234	230	229	[illegible]	[illegible]	79	160	160	209	[illegible]	[illegible]	[illegible]	[illegible]	133	133	171	246	240
1930	1	[illegible]	236	[illegible]	[illegible]	[illegible]	[illegible]	78	161	161	209	208	174	77	97	131	132	173	236	[illegible]
2000	1	230	230	225	224	235	236	79	155	157	206	206	[illegible]	75	97	130	131	[illegible]	[illegible]	[illegible]

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Remarks

Report No.

PWA FID REV 3/71

Pratt & Whitney Aircraft

U
A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. A

Date 1/25/78

Engineer Bill Gamble

Operators Corky / Renaudi

D-4 Stand Engine/Rig No. E34024 Build 10 Project

Type of Test 50 HR. ENDURANCE RUN

Time A.M. P.M.	Tot. Idles / Mission Point No	T17 Forw Dome Air A-20	T18 Rear Dome Air A-21	T19 G/B OIL IN A-22	T20 G/B OIL OUT A-23		B-11 No. 2 BRG. B-1	B-12 No. 2 BRG. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft BRG B-12	T21 G/B BRG #1 A-24	T22 G/B BRG #2 A-25
0815	Z	69	69	64	68		71	—	71	—	70	71	71	71	71	69	71	71	69	68
0900		START ROTATION						—												
0935		ON ENDURANCE						—												
0940		76	76	81	94		228	—	251		233	228	236	236	125	122	242	243	120	103
1010		OFF ENDURANCE						—												
1020		380	381	93	125		253	—	279	—	254	247	265	259	290	161	273	271	137	150
1025	2	ON ENDURANCE																		
1030		473	478	96	131		303	—	329	—	305	305	298	314	325	165	277	273	141	155
1040		OFF ENDURANCE																		
1045		STOP																		
1445		START																		
1500	1	ON ENDURANCE																		
1505		83	208	85	98	1	223	—	240	—	226	221	230	230	252	160	237	235	103	106
1530		82	83	85	99		243	—	260	—	244	239	247	246	267	171	252	212	103	107
1600		87	82	86	99		222	-	241	-	224	219	227	228	249	163	234	235	102	107
1630		81	81	85	99		216	-	234	—	217	211	220	220	242	161	226	221	103	105
1700		80	80	79	86		236	-	249	—	231	228	235	233	255	166	236	237	103	107
1730		78	78	85	99		224		237	—	218	214	222	221	244	161	224	225	102	106
1800		78	77	85	98		241	—	260	—	239	235	242	240	260	168	240	241	102	105
1830		78	78	85	98		237	-	256	—	239	234	242	263	263	170	249	250	103	103
1900		77	77	85	98		243	—	260	—	243	239	246	246	267	171	251	252	102	106
1930		76	76	85	98		244	—	261	—	244	239	247	247	267	171	252	253	101	105
2000		77	77	84	97		242	-	259	-	242	237	245	244	265	170	249	250	101	103

Remarks

Page No.

Report No.

PWA

Pratt & Whitney Aircraft

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LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet: 1/4 0

Date 01-25-78

Engineer B. Gamble

Operators Reid/Giles

D-4 Stand [redacted]/Rig No. 34024 Build 10 Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time Act. P.M.	Tot. Hrs / Mission Point	P_{21} LAB Cooling (PSIG) G2	P10 Forward dome (PSIG) G11	P_{11} REAR Dome G12	P_3 2,3 OIL (PSIA) G13	P_4 1,4,5 OIL G14	P_2 OIL Pump G15	P14 Bore Exit G16	P_{11} G/B OIL IN (PSIG) G17	P_{20} G/B OIL out G18	P_5 2,3 Breather (PSIA) G19	P_9 Bore Chamb G20	P_7 Forward Chamber G21	P_8 Rear Chamb G22	P_{15} Breather orifice G37	P_{16} 2,3 G38	P_{17} Forward Dome orifice G39	P_{18} Rear Dome orifice G40	P_1 OIL TANK G41	P6 1,4,5 AIR G42	ΔPN ±	ΔP16 ±
2030	1	25.5	2.0	6.0	43	28	109	15	14	0	14	18.5	15.5	17.5	13.7	15.7	18	24	13.5	66	.1	0
2100		25.5	2.0	6.0	43	28	109	15	14	0	14	18	15.5	17.5	13.7	15.7	18	24	13.5	66.5	.1	0
2130		25.5	2.0	6.0	43	28	109	15	14	0	14	18.5	15.5	17.5	13.7	15.7	18	24	13.5	67.5	.1	0
2200		25.5	2.0	6.0	43	28.5	108.5	15	14	0	14	18.5	15.5	17.5	13.7	15.7	18	24	13.5	67.5	.1	0
2220		25.5	2.0	6.0	43	28.5	107	15	14	0	14	19	15.5	17.5	13.7	15.7	18	24	13.5	67	.1	0
2240		25.5	2.0	6.0	43	28.5	110	15	14	0	14	18.5	15.5	17.5	13.7	15.7	18	24	13.5	67	.1	0
2245	1	OFF ENDURANCE																				
2250		Stop																				
		1/26/78			1st shift																	
0800	2	0	0	0	14.7	14.7	14.7	14.7	0	0	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	0	0
0815		Start Rotation																				
0830	6	ON Endurance																				
0835	1	26	2.0	6.0	44	29	110	15	14	0	14.2	18.0	15.5	17.0	13.6	15.0	18.0	23.5	14.0	69	.1	0
0900		26	2.0	6.0	45	29.5	121	15	14	0	14.2	18.0	15.5	17.5	13.6	15.7	18.0	23.5	14.0	69	.1	0
0930		27	2.0	6.0	44	28.0	111	15	14	0	14.2	18.0	15.5	17.0	13.7	15.8	18.0	24.0	14.0	65	.1	0
1000		26	2.0	6.0	44	28.0	114	15	14	0	14.2	19.0	15.5	17.0	13.6	15.8	18.0	24.0	14.0	66	.1	0
1030		26	2.0	6.0	44	28.0	111	15	14	0	14.2	19.0	15.5	17.0	13.6	15.8	18.0	24.0	14.0	66.5	.1	0
1100		26	2.0	6.0	44	28.0	112	15	14	0	14.2	19.0	15.5	17.0	13.6	15.7	18.0	24.0	14.0	66.0	.1	0
1130		26	2.0	6.0	44	29.0	112	15	14	0	14.2	19.0	15.5	17.0	13.6	15.8	18.0	24.0	14.0	67.0	.1	0
1200		26	2.0	6.0	44	29.0	112	15	14	0	14.2	19.0	15.5	17.0	13.6	15.8	18.0	24.0	14.0	67.0	.1	0
1230		26	2.0	6.0	44	29.0	113	15	14	0	14.2	19.0	15.5	17.0	13.6	15.8	18.0	24.0	14.0	67.0	.1	0

Remarks

Point #1, Endur. 2245 / 2000 / 2:45 hr RT 2250 / 2000 / 2:50 hr

Point # 6; Endur. 1230 / 0830 / 4:00 hr RT 1230 / 0815 / 4:15 hr

Page No. Report No.

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Pratt & Whitney Aircraft — U.A.
FLORIDA RESEARCH AND DEVELOPMENT CENTER

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 2/40
Date 01-25-78
Engineer [illegible] Gamble
Operators Reid / Giles

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs / MISSION POINT NO.	"U" Tubes Dp 6 (INCHES HG) U-63	Dp 15 U-64	Dp 17 U-65	Dp 18 U-66		FLOW METERS F1 OIL Supply PPM	F2 1,4,5 OIL PPM	F3 2,3 OIL PPM	F4 G/B PPM	Pump Speed RPM	Rig Speed RPM	VIBRATIONS (MILS) RIG NO. 3 BRG 350° V-1	RIG NO. 3 BRG 260° V-2	RIG NO. 2 BRG VERT V-3	RIG FRONT VERT V-4	RIG FRONT HORZ V-5	RIG REAR VERT V-6	RIG REAR HORZ V-7	G/B VERT V-8	G/B HOR V-9
2030		9.0	.3	.6	4.6		116.7	54.4	62.1	11.5	6623	9080	.14	.16	.1	.15	.21	.05	.20	.22	.25
2100		9.0	.3	.6	4.6		116.2	53.9	62.2	11.5	6642	9090	.14	.17	.1	.15	.21	.06	.20	.22	.25
2130		9.3	.3	.6	4.7		116.5	53.9	62.0	11.5	6620	9100	.14	.17	.1	.15	.22	.06	.20	.22	.25
2200		9.3	.3	.6	4.7		116.2	53.5	62.5	11.5	6634	9090	.14	.17	.1	.15	.22	.06	.20	.22	.25
2220		9.3	.3	.6	4.7		115.8	53.3	62.4	11.5	6638	9100	.14	.17	.1	.15	.22	.06	.20	.22	.25
2240		9.3	.3	.6	4.7		117.0	54.4	62.4	11.5	6631	9080	.14	.17	.1	.15	.22	.07	.20	.23	.25
2245		OFF		ENDURANCE																	
2250		Stop																			
				1/26/78		W/Shift															
0800	2	0	0	0	0		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0815		Start	Rotation																		
0830	4	ON	Endurance																		
0835		10.3	.3	.5	4.6		117.2	53.8	63.3	11.2	6608	9050	.15	.19	.25	.16	.27	.07	.22	.21	.22
0900		10.5	.3	.5	4.5		117.5	54.2	62.5	11.3	6566	8990	.13	.16	.23	.15	.23	.09	.01	.22	.21
0930		9.6	.4	.7	4.6		116.0	53.9	62.0	11.4	6597	9040	.15	.18	.25	.15	.25	.08	.01	.22	.22
1000		9.1	.3	.6	4.5		118.0	55.0	62.9	11.3	6648	9100	.15	.20	.27	.15	.3	.07	.01	.21	.22
1030		9.1	.4	.5	4.5		118.5	55.4	62.9	11.4	6666	9140	.16	.20	.27	.15	.28	.07	.21	.21	.22
1100		9.0	.4	.5	4.5		118.5	53.3	63.0	11.4	6667	9140	.16	.20	.27	.15	.29	.07	.21	.21	[illegible]
1130		9.5	.4	.5	4.4		117.1	54.4	63.3	11.4	66[illegible]	9170	.16	.20	.27	.15	.28	.07	.22	.22	.22
1200		[illegible]	.4	.5	4.5		[illegible]	[illegible]	62.9	11.4	6674	9140	.16	.20	.27	.15	.29	.07	.22	.21	.22
1230		9.6	.4	.5	4.5		117.0	53.1	62.8	11.4	6695	9170	.16	.20	.27	.15	.28	.07	.22	.21	.22

Remarks

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Pratt & Whitney Aircraft

U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. [illegible]

Date 1-25-78

Engineer Bill Gamble

Operators Coney/Bennett

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN

"A" Switch OF

Time A.M./P.M.	Test Hrs / Mission Point No.	T_1 Oil Tank	T_2 Oil Pump Dis.	T_3 2/3 Comp Sup	T_4 1-4-5 Oil Sup.	T_{51} 2/3 Comp Air	T_{52} 2/3 Comp Air	T_6 1-4-5 Air Ori.	T_{71} Forw Cham Temp ×	T_{72} Forw Cham Temp	T_{81} Rear Cham Temp ×	T_{82} Rear Cham Temp	T_9 Bore Temp ×	T_{10} Forw Dome Temp	T_{12} Forw Dome Temp	T_{11} Rear Dome	T_{13} Rear Dome	T_{15} Brea. Air Orif.	T_{16} 2/3 Scav. Pum Disc	T_{14} Bore Air Out
		A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9	A-10	A-11	A-12	A-13	A-14	A-15	A-16	A-17	A-18	A-19
2030	1	219	219	211	212	224	225	78	157	157	205	205	185	98	97	131	131	[illegible]	232	141
2100		223	223	217	217	229	230	75	156	156	204	204	184	96	95	128	129	166	240	127
2130		225	225	219	219	230	231	77	154	154	203	203	185	96	95	128	129	165	238	136
2200		236	235	220	219	238	237	77	160	160	208	208	226	97	96	131	132	170	239	150
2220		239	238	232	231	243	243	76	141	160	209	209	249	97	96	131	132	171	215	157
2240		218	218	211	210	224	225	75	156	156	205	205	246	96	95	130	131	166	231	151
2245		OFF		ENDURANCE																
2250		Stop																		
		1st SHIFT					1-26-78													
0800	2	64	59	64	108	57	57	104	102	97	97	59	102	99	93	87	59	59	105	61
0815			START																	
0830	6																			
[illegible]		210	208	205	204	214	215	68	131	131	182	182	202	73	74	110	110	124	233	115
0900		202	202	194	206	204	205	61	141	140	189	189	229	78	79	115	116	141	204	119
0930		216	215	198	197	219	218	63	147	146	193	193	223	82	83	118	119	149	219	133
1000		213	213	206	205	219	219	64	149	149	197	196	199	83	84	121	121	151	226	137
1030		213	212	201	200	217	217	66	149	148	197	196	183	85	86	122	122	152	222	131
1100		211	210	205	204	216	217	66	148	148	196	197	187	85	87	123	123	152	224	133
1130		222	221	218	217	229	229	67	150	150	199	199	189	85	87	123	124	152	[illegible]	133
1200		219	219	205	223	223	224	67	152	151	200	200	190	87	88	124	124	154	226	133
1230		219	218	199	[illegible]	[illegible]	[illegible]	68	153	153	201	200	192	88	88	[illegible]	124	[illegible]	[illegible]	[illegible]

Remarks

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Report No.

PWA [illegible]

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

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LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 4/11
Date 1-25-78
Engineer BILL GAMBLE
Operators CONLEY/BERNARD

D-4 Stand   Engine/Rig No. E34024   Build 10   Project ______

Type of Test 50 HR. ENDURANCE RUN

A.M. P.M. (Time)	Tot. Hrs	T17 Forw. Dome Air A-20	T18 Rear Dome Air A-21	T19 G/B Oil In A-22	T20 G/B Oil Out A-23	B-11 No. 2 BRG. B-1	B-12 No. 2 BRG. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft BRG B-12	T21 G/B Brg. #1 A-24	T22 G/B Brg #2 A-25
2030	1	76	76	84	97	232		250		233	227	236	237	257	166	242	243	101	105
2100		74	74	84	96	241		259		243	237	245	244	265	169	249	250	100	104
2130		75	76	84	96	238	-	257		240	234	242	242	263	168	248	249	100	103
2200		76	76	84	97	238		256		239	231	242	242	262	167	245	248	101	103
2220		75	75	84	97	245		262		246	241	248	248	268	169	253	254	101	103
2240		74	74	84	97	231		249		232	227	235	236	257	164	242	243	101	105
2245		OFF	INDURANCE																
2332		Stop																	
			1st	SHIFT			1-26-78												
[illegible]	2	58	58	65	68	108	—	109	—	106	107	105	105	106	82	110	110	67	67
[illegible]			START																
0830	6		IN	END															
0835		61	61	77	87	223	—	243	—	226	230	228	229	251	151	235	234	92	97
0900		62	62	79	90	184	—	201	—	181	176	184	184	211	134	189	191	95	99
0930		44	64	81	92	206		234		217	211	220	221	243	151	226	227	96	100
1000		64	64	81	93	224		240		215	214	221	219	243	151	223	224	97	99
1030		67	66	81	93	221		239		222	223	217	226	227	155	233	234	98	102
1100		67	67	82	94	224		143		223	218	226	224	248	156	230	231	98	101
1130		68	68	83	[illegible]	226		251		230	223	229	228	257	157	230	231	98	102
1200		68	68	81	93	225		244		227	221	230	229	251	157	235	237	98	102
1230		69	69	82	[illegible]	216		235		218	212	223	224	244	154	228	227	99	[illegible]

Remarks

Page

Pratt & Whitney Aircraft

U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 1/4 C

Date 1/20/82

Engineer Bill Gamble

Operators Cutler / Brown
2nd Reid / Giles

D-4 Stand [redacted]/Rig No. 34024 Build 10 Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time A.M. P.M.	Tot. Hrs (Mission Point)	P_{21} LAB Cooling	P_{10} Forward dome	P_{11} REAR Dome	P_3 2,3 OIL	P_4 1,4,5 OIL	P_2 OIL Pump	P_{14} Bore Exit	P_{19} G/B OIL IN	P_{20} G/B OIL out	P_5 2,3 Breather	P_9 Bore Chamb	P_7 Forward Chamber	P_8 Rear Chamb	P_{15} Breather orifice	P_{16} 2,3 Scavenge	P_{17} Forward Dome orifice	P_{18} Rear Dome orifice	P_1 OIL TANK	P_6 1,4,5 AIR	ΔP
		← PSIG		→ ←	PSIA			→ ← PSIG		→ ←				PSIA						→	
		G2	G11	G12	G13	G14	G15	G16	G17	G18	G19	G20	G21	G22	G37	G38	G39	G40	G41	G42	
1300	4	26	2.0	6.0	44	29	111	15	14	0	14.5	18	15.5	17	13.6	16	18	24	14	67	.10
1330		26	2.0	6.0	44	29	113	15	14	0	14.2	19	15.5	17	13.7	16	18	24	14	68	.10
1400		26	2.0	6.0	43.5	29	113	15	14	0	14.2	18.5	15.5	17	13.6	16	18	24	14	68	.10
1430		26	2.0	6.0	44	29	115	15	14	0	14.5	18	15.5	17	13.6	16	18	24	14	67	.10
1500		26	2.0	6.0	44	29	114	15	14	0	14.2	18	15.5	17	13.6	16	18	24	14	68	.10
1530		26	2.0	6.0	44	29	112	15	14	0	14.5	19	15.5	17	13.7	16	18	24	14	67.5	.10
1600		25	2.0	6.0	44	29	106	15	14	0	14.1	18	15.5	17	13.6	15.9	18	23.5	14	67	.10
1620		26	2.0	6.0	44	29	110	15	14	0	14	18	15.5	17	13.7	15.9	18	23.5	14	67	.10
1640		27	2.0	6.0	44	29	110	15	14	0	14	18.5	15.5	17	13.7	15.9	18	24	14	68.5	.10
1646	6	OFF		ENDURANCE																	
1730	3	ON		ENDURANCE																	
1730		30	14	18	51	32	131	16	14	0	6.8	25	18.5	24.5	3.6	14.0	30	39	10	76.5	.50
1800		30	13	18	51	32	132	15.5	14	0	6.7	25	18.0	24.5	3.6	13.9	26.5	40	9.9	76.5	.40
1830		30	12	20	51	32	133	16	14	0	6.7	25	18	25	3.9	13.9	26	43	9.8	76.0	.50
1900		30	11	20	51	32	132	15.5	14.5	0	6.8	25	18	24	3.9	13.8	26.5	40	9.8	76.0	.50
1930		30	11	20	51	32	133	15.5	14	0	6.8	25	18	25	3.9	13.8	26	40.5	9.8	76.5	.50
2000		30	11	20	51	32	130	15.5	14	0	6.8	25	18	25	3.9	13.9	25	42.5	9.8	78	.50
2030		30	11	20	51	32	133	15.5	14	0	6.8	25	18	25.5	3.9	13.9	25	42	9.8	78	.50
2100		30	11	20	51	32	131.5	16	14	0	6.8	25.5	18	25	3.9	13.9	25	42	9.8	78	.50
2130		30	11	20	51	32	132.5	16	14	0	6.8	25	18	25	3.9	13.9	27	40	9.8	78	.50
2200		30	11	20	51	32	132.5	16	14	0	6.8	25	18	25	3.9	13.9	26	40.5	9.8	78	.50
2230		30	11	19.5	51	32	132	16	14	0	6.8	25	18	25	3.9	13.9	27.5	40.5	9.8	78	.50
2300		30	11	21	51	32	133	16	14	0	6.7	25	18	25.5	4.0	13.9	27	43	9.8	78	.50

Page No.

Remarks

Report No.

PWA FORM 48959

Pratt & Whitr y Aircraft U A.

LOG OF F"GINE TEST

EXPERIMENTA ‹EST DEPARTMENT

Sheet No. 2/7C

Date 1/26/78

Engineer [illegible] Gamble

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Operators Cagley/Barnes 2nd Reid/Giles

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs (Mission Point No.)	"U" Tubes Dp 6 (Inches HG) U-63	Dp 15 U-64	Dp 17 U-65	Dp 18 U-66		Flow Meters F1 Oil Supply PPM	F2 1,4,5 Oil PPM	F3 2,3 Oil PPM	F4 G/B PPM	Pump Speed RPM	Rig Speed RPM	Vibrations (Mils) R/C No. 3 Brg 350° V-1	R/C No. 3 Brg 260° V-2	R/C No. 2 Brg Vert V-3	R/C Front Vert V-4	R/C Front Horz V-5	R/C Rear Vert V-6	R/C Rear Horz V-7	G/B Vert V-8	G/B Hor -9
1300	6	9.5	.4	.5	4.5		118.2	54.9	63.3	11.4	6659	9130	.16	.20	.27	.15	.27	.07	.22	.21	.22
1330		9.8	.4	.6	4.6		117.9	54.7	63.1	11.5	6627	9070	.16	.22	.27	.15	.28	.07	.22	.21	.22
1400		9.7	.4	.5	4.6		117.8	54.7	63.1	11.4	6616	9070	.16	.20	.27	.15	.28	.07	.22	.21	.22
1430		9.5	.4	.6	4.6		118.6	54.9	63.2	11.5	6683	9150	.17	.20	.27	.15	.28	.07	.22	.21	.25
1500		9.9	.4	.6	4.6		118.7	55.2	63.3	11.5	[illegible]	9130	.15	.20	.25	.15	.26	.07	.22	.21	.22
1530		9.9	.4	.6	4.6		118.3	54.9	63.3	11.5	6664	9130	.16	.20	.26	.15	.27	.07	.22	.21	.22
1600		9.4	.4	.6	4.6		119.8	56.4	63.1	11.5	6670	9120	.15	.19	.25	.15	.25	.07	.20	.22	.24
1620		9.5	.4	.6	4.6		117.1	53.6	63.2	11.5	6718	9200	.16	.19	.26	.16	.27	.08	.21	.20	.23
1640		9.5	.4	.6	4.6		116.6	52.6	63.5	11.5	6700	9190	.16	.20	.27	.16	.28	.07	.21	.20	.23
1646	6	OFF ENDURANCE																			
1730	3	ON ENDURANCE																			
1730		11.0	1.2	1.6	12.3		134	61.5	72.0	11.6	7987	10930	.07	.09	.17	.12	.11	.12	.13	.1	.34
1800		11.0	1.2	1.3	13.2		134.4	61.7	71.9	11.6	7983	10940	.07	.12	.14	.11	.09	.10	.20	.10	.34
1830		11.0	1.2	1.3	14.8		134.9	62.4	71.9	11.7	7990	10940	.07	.13	.14	.11	.09	.10	.03	.10	.34
1900		11.0	1.2	1.3	13.1		134.2	61.6	71.8	11.7	8008	10960	.06	.11	.15	.1	.08	.1	.01	.09	.32
1930		11.0	1.2	1.3	13.5		134.9	62.3	72.0	11.6	7991	10850	.07	.12	.14	.1	.08	.1	.02	.09	.33
2000		11.2	1.2	1.3	14.4		134.3	61.5	71.9	11.7	7963	10910	.08	.13	.14	.12	.08	.09	.02	.09	.33
2030		11.0	1.2	1.3	14.3		134.2	61.7	71.9	11.7	7944	10910	.09	.15	.13	.12	.07	.08	.02	.10	.34
2100		11.0	1.2	1.3	14.1		134.3	61.5	72.0	11.7	7996	10940	.09	.14	.13	.12	.08	.08	.02	.10	.31
2130		11.0	1.2	1.3	13.1		135.0	62.1	72	11.6	8000	10960	.09	.12	.13	.11	.08	.09	.02	.10	.32
2200		11.0	1.2	1.3	13.6		134.5	61.9	72.2	11.6	7981	10920	.07	.13	.13	.11	.07	.09	.02	.10	.32
2230		11.0	1.2	1.3	13.4		134.5	61.9	71.8	11.6	7990	10910	.07	.11	.14	.11	.09	.10	.01	.10	.34
2300		11.0	1.2	1.3	15.5		134.5	61.9	72.1	11.6	7956	10900	.07	.12	.14	.05	.08	.10	.02	.11	.35

Remarks Oil Added. (1½ qt @ 1730) (½ gal @ 1900) (½ gal @ 2000) (½ gal @ 2100)
½ gal @ 2230

Page No.

FD 4899

Pratt & Whitney Aircraft

U
A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet C

Date 3/26/48

Engineer Bill Ga[illegible]

Operators Conley/Bernard; 2nd Rem Giles

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN

Time A.M. P.M.	Tot. Hrs / Mission Point No.	T1 Oil Tank A-1	T2 Oil Pump Dis. A-2	T3 2/3 Comp Sup A-3	T4 1-4-5 Oil Sup. A-4	TS1 2/3 Comp Air A-5	TS2 2/3 Comp Air A-6	T6 1-4-5 Air ORI. A-7	T7₁ Forw Cham Temp A-8	T7₂ Forw Cham Temp A-9	T8₁ Rear Cham Temp A-10	T8₂ Rear Cham Temp A-11	T9 Bore Temp A-12	T10 Forw Dome Temp A-13	T12 Forw Dome Temp A-14	T11 Rear Dome A-15	T13 Rear Dome A-16	T15 Brea. Air ORIF. A-17	T16 2/3 Scav. Pum Disc A-18	T14 Bore Air Out A-19
		"A" Switch OF																		
1300	10	208	208	203	203	214	215	69	146	146	196	196	188	87	89	124	124	149	223	152
1330		206	206	198	198	211	211	68	148	148	197	196	207	89	90	125	126	151	217	141
1400		205	205	198	197	211	211	70	149	149	198	198	212	89	91	126	127	151	217	149
1430		208	207	201	201	214	214	70	148	148	199	199	229	89	91	127	128	149	221	151
1500		204	204	193	[illegible]	210	210	70	148	148	197	[illegible]	175	89	91	[illegible]	[illegible]	[illegible]	225	[illegible]
1530		[illegible]	207	201	201	214	214	72	148	149	199	202	202	96	92	128	128	152	222	144
1600		202	202	187	186	208	208	71	151	151	201	201	236	92	92	127	128	153	208	131
1620		227	227	222	221	233	233	70	157	157	206	205	233	90	92	128	128	158	240	136
1640		236	236	231	230	241	242	72	159	159	208	208	282	92	93	130	130	163	246	163
1646	6	OFF	ENDURANCE																	
1730	3	ON	ENDURANCE																	
1730		262	262	250	250	266	267	73	239	238	356	357	351	215	215	301	302	222	213	268
1800		259	259	247	246	263	264	73	231	231	351	351	339	197	198	295	297	214	268	260
1830		253	253	241	240	257	258	73	223	222	359	359	340	190	191	309	311	211	264	267
1900		261	261	249	248	265	266	73	231	230	355	355	335	197	197	299	300	214	271	262
1930		258	259	247	246	263	264	73	228	227	351	352	332	193	194	296	298	211	269	259
2000		[illegible]	[illegible]	240	239	[illegible]	257	73	[illegible]	224	357	357	330	183	184	302	304	[illegible]	265	260
2030		255	255	243	242	259	261	72	221	220	355	355	337	183	183	302	305	215	265	264
2100		264	264	252	251	267	268	72	224	223	357	358	338	184	184	303	304	216	275	265
2130		262	262	251	250	266	267	71	229	229	353	354	335	194	194	296	298	214	271	262
2200		259	259	247	246	263	264	70	223	223	350	350	329	186	186	294	296	212	270	255
2230		259	259	247	246	263	264	69	231	230	350	350	331	199	199	294	296	210	269	258
2300		261	261	248	247	265	267	69	228	227	359	360	[illegible]	173	174	309	311	211	270	267

Remarks

Point #6 1646 / 1230; Endur: 416; 1646 / 1230

Point # 3 2300 / 1730; Endur 530; 2300 / [illegible]

Page No.

Report No.

Pratt & Whitney Aircraft
Florida Research and Development Center

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 14C
Date 01/26/78
Engineer BILL GAMBLE
Operators Cauley/Bernard
2nd Reid/Giles

D-4 Stand Engine/Rig No. E34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN

"A" SW OF "B" SWITCH OF

Time A.M./P.M.	Tot. Hrs / Mission No. / Event No.	T17 Forw Dome AIR OC(F) A-20	T18 Rear Dome Air A-21	T19 G/B OIL IN A-22	T20 G/B OIL out A-23	B-11 No.2 BRG. B-1	B-12 No.2 BRG. B-2	B-21 No.3 Brg. B-3	B-22 No.3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft BRG B-12	T21 G/B BRG #1 A-24	T22 G/B BRG #2 A-25
1300	6	71	71	83	94	222		242		225	219	227	228	250	158	234	235	99	103
1330		71	71	82	94	217		236		217	212	220	219	243	151	225	226	99	101
1400		72	72	82	95	216		236		218	213	221	221	244	155	228	229	99	103
1430		72	72	82	95	219		238		218	212	219	218	241	153	221	220	100	103
1500		72	73	82	94	217		237		218	213	221	221	244	153	[illegible]	[illegible]	99	102
1530		73	74	83	95	221		241		224	218	227	227	249	157	232	234	100	102
1600		72	72	83	95	201		218		200	194	203	202	227	148	207	209	100	105
1620		72	72	83	95	241		260		243	238	246	246	266	165	251	251	100	104
1640		73	73	83	95	239		258		241	235	244	244	264	164	250	250	100	103
1646	6	OFF ENDURANCE																	
1730	3	ON Endurance																	
1730		218	256	87	107	274		295		271	265	276	272	303	169	282	282	117	117
1800		187	254	88	108	268		290		266	261	271	267	298	170	277	277	117	117
1830		187	274	88	110	263		287		262	257	269	264	275	168	271	275	118	119
1900		190	258	88	109	269		291		267	262	273	269	279	171	278	279	117	117
1930		[illegible]	257	87	108	268		291		266	261	272	268	299	170	277	278	116	117
2000		173	265	87	[illegible]	263		287		262	257	269	265	295	168	275	276	116	117
2030		173	266	87	109	264		287		263	258	269	265	276	165	274	275	117	119
2100		171	264	88	109	274		297		273	269	280	276	305	172	286	286	117	119
2130		185	253	87	108	271		296		271	266	276	272	303	170	279	279	117	117
2200		177	257	86	106	269		292		268	263	273	270	300	169	276	280	114	115
2230		193	254	86	108	269		290		266	261	271	267	298	167	277	277	115	117
2300		188	277	86	107	269		292		268	262	274	270	300	167	280	281	117	122

Remarks

Page No. Report No.

Pratt & Whitney Aircraft U.A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 14 D

Date 01-26-78

Engineer Bill Gamble

Operators Reed/Giles

D-4 Stand ~~Engine~~/Rig No. 34024 Build 10 Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time A.M./P.M.	Tot. Hrs. / Mission Point No.	P_{21} LAB Cooling (PSIG) G.2	P_{10} Forward dome (PSIG) G11	P_{11} REAR Dome G12	P_3 2,3 OIL (PSIA) G13	P_4 1,4,5 OIL G14	P_2 OIL Pump G15	P_{14} Bore Exit G16	P_{19} G/B OIL IN (PSIG) G17	P_{20} G/B OIL out G18	P_5 2,3 Breather (PSIA) G19	P_9 Bore Chamb G20	P_7 Forward Chamber G21	P_8 Rear Chamb G22	P_{15} Breather orifice G37	P_{16} 2,3 Scavenge G38	P_{17} Forward Dome orifice G39	P_{18} Rear Dome orifice G40	P_1 OIL TANK G41	P_6 1,4,5 AIR G42	ΔP_2 ΔP_{16}
2330	3	30	11	21	51	32	132	16	14	0	6.8	26	18	26	4.0	13.9	27	45	9.9	78	510
2330	3	OFF ENDURANCE																			
2335		STOP																			

Remarks: Point # 3 2330 / 2300 / Chour: :30

ET: 2335 / 2300 / :35

Page No. Report No.

FRDC 073 REV 7/71 48959

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

U
A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 2/10
Date 01-26-78
Engineer [illegible] Gamble
Operators Reid/Giles

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time		"U" Tubes					FLOW METERS						VIBRATIONS								
A.M. P.M.	Tot. Hrs. MISSION POINT NO.	Dp 6	Dp 15	Dp 17	Dp 18		F1 OIL Supply	F2 1,4,5 OIL	F3 2,3 OIL	F4 G/B	Pump speed	Rig speed	RIG NO. 3 BRG 350°	RIG NO. 3 BRG 260°	RIG NO. 2 BRG VERT	RIG FRONT VERT	RIG FRONT HORZ	RIG REAR VERT	RIG REAR HORZ	G/B VERT	G/B HOR.
		INCHES HG					PPM	PPM	PPM	PPM	RPM	RPM	MILS								
		U-63	U-64	U-65	U-66								V-1	V-2	V-3	V-4	V-5	V-6	V-7	V-8	-9
2330		11.0	1.2	1.3	15.8		134.1	61.3	72.0	11.6	7942	10870	.07	.12	.13	.12	.09	.10	.02	.12	.36
2330	3	OFF ENDURANCE																			
2335		STOP																			

Remarks

Page No.

FD 4866

Pratt & Whit[illegible] Aircraft U A.

LOG OF E[illegible]GINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet [illegible] D

Date 01-26-78

Engineer BILL GAMBLE

Operators

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN

Time A.M. P.M.	Test Hrs / MISSION POINT NO.	T1 OIL TANK A-1	T2 OIL Pump Dis. A-2	T3 2/3 Comp Sup A-3	T4 1-4-5 OIL Sup. A-4	TS1 2/3 Comp Air A-5	TS2 2/3 Comp Air A-6	T6 1-4-5 Air ORI. A-7	T71 Forw Cham Temp A-8	T72 Forw Cham Temp A-9	T81 Rear Cham Temp A-10	T82 Rear Cham Temp A-11	T9 Bore Temp A-12	T10 Forw Dome Temp A-13	T12 Forw Dome Temp A-14	T11 Rear Dome A-15	T13 Rear Dome A-16	T15 Brea. Air ORIF. A-17	T16 2/3 Scav. Pum Disc A-18	T14 Bore Air Out A-19
		"A" SWITCH °F																		
2330		262	262	249	248	266	267	67	231	230	368	368	348	197	198	319	321	215	273	270
2330	3		OFF	ENDURANCE																
2335			STOP																	

Remarks

Page No.

Report N.

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

U
A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. ______

Date 01-26-78

Engineer BILL GAMBLE

Operators ______

D-4 Stand ~~Engine~~/Rig No. F34024 Build 10 Project ______

Type of Test 50 HR. ENDURANCE RUN

Time	"A" SW OF					"B" SWITCH OF														
A.M. P.M.	Tot. Hrs MISSION POINT NO.	T17 Frwd Dome Air A-20	T18 Rear Dome Air A-21	T19 G/B OIL IN A-22	T20 G/B OIL out A-23		B-11 No. 2 BRG. B-1	B-12 No. 2 BRG. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft BRG B-12		T21 G/B BRG #1 A-24	T22 G/B BRG #2 A-25
2330		193	288	85	106		273	—	295	—	271	267	278	274	303	168	284	284		116	120
2330	3		OFF	ENDURANCE																	
2335			STOP																		

Remarks

Page No.

Report No.

Pratt & Whitney Aircraft   U A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet 19 E
Date 1-26-78
Engineer B. Gai[illegible]
Operators Reed/Giles

D-4 Stand ____ /Rig No. 34024 Build 10 Project ____

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time	Tot. Hrs (Mission Point No.)	P_{21} LAB Cooling	P_{10} Forward dome	P_{11} REAR Dome	P_3 2,3 OIL	P_4 1,4,5 OIL	P_2 OIL Pump	P14 Bore Exit	P_{19} G/B OIL IN	P_{20} G/B OIL out	P_5 2,3 Breather	P_9 Bore Chamb	P_7 Forward Chamber	P_8 Rear Chamb	P_{15} Breather orifice	P_{16} 2,3 scavenge	P_{17} Forward Dome orifice	P_{18} Rear Dome orifice	P_1 OIL Trans	P_6 1,4,5 AIR	ΔP_{16}
		PSIG	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	
		G2	G11	G12	G13	G14	G15	G16	G17	G18	G19	G20	G21	G22	G37	G38	G39	G40	G41	G42	
1330	3	30	11	21	51	32	132	16	14	0	6.2	26	18	26	4.0	13.9	27	45	9.9	72	5.0
2220	3	OFF Endurance																			
2225		S/D																			
1715	3	Start		1/27/78																	
1730		ON ENDURANCE																			
1730		30	11	19	52	33	138	16	14	0	2.9	25.5	18	24.5	3.7	14.5	25.5	39	10.2	79	5.0
1800		30	10	21	51	32	133	16	14	0	6.8	25.5	18	25.5	3.8	14.0	23	44	10	77	4.0
1830		30	10	21	51	32	131	16	14	0	6.8	25	18	25.5	3.8	14.0	26	42.5	9.9	77	5.0
1900		30	10	20	51	32	135	16	14	0	6.7	25	18	25	3.7	14	26	40	9.9	77	5.2
1930		30	10	20	51	32	134	16	14	0	6.6	25	18	25	3.7	14	26	40	10	77	5.0
2000		30	10	20	51	32	133	16	14	0	6.6	25	18	25	3.8	14	27	40	10	77	5.0
2030		30	10	21	51	32	134	16	14	0	6.7	25	18	25	3.8	14	25	43	10	77	5.0
2100		30	10	21	51	32	133	16	14	0	6.6	25	18.5	25	3.9	14	32	43	10	77	5.0
2130		30	10	20	51	32	130	16	14	0	6.6	25	18	25	3.9	14	26	40	10	77	5.0
2200		30	10	20	51	32	132	16	14	0	6.8	25	18	25	4.0	14	26	43	10	76	5.0
2230		30	10	20	51	32	133	16	14	0	6.7	25	18	25	3.8	14	27	41	10	76	5.0
2300		30	10	20	50	32	131	16	14	0	6.6	25	18	25	3.8	14	27	41	10	77	5.0
2330		30	10	20	50	32	133	16	14	0	6.6	25	18	25	3.8	14	27	42	10	77	5.0
2330	3	OFF ENDURANCE																			
2335		SHUT DOWN																			

Remarks: Point #3 [illegible]

Point #3 ENDUR: 2330 − 1730 = 6:00 at 2335 − 1715 = 6:20

Pratt & Whitney Aircraft — Florida Research and Development Center

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 2/4E
Date 01-26-78
Engineer [illegible]
Operators [illegible]

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs / Mission Point No.	"U" Tubes Dp 6 (inches HG) U-63	Dp 15 U-64	Dp 17 U-65	Dp 18 U-66		Flow Meters F1 Oil Supply PPM	F2 1,4,5 Oil PPM	F3 2,3 Oil PPM	F4 G/B PPM	Pump Speed RPM	Rig Speed RPM	Vibrations (MILS) Rig No. 3 Brg 350° V-1	Rig No. 3 Brg 260° V-2	Rig No. 2 Brg Vert V-3	Rig Front Vert V-4	Rig Front Horz V-5	Rig Rear Vert V-6	Rig Rear Horz V-7	G/B Vert V-8	G/B Hor V-9
2330		11.?	1.2	1.3	15.4		134.1	61.3	72.0	11.6	7942	10870	.07	.12	.13	.12	.09	.10	.02	.12	.36
2330		Off Endurance																			
2335		S.D.																			
1715	3	Start	1/27/78																		
1730		On Endurance																			
1730		11.0	1.2	1.0	12.0		134.8	61.7	72.2	11.3	7934	10980	.09	.09	.15	.14	.11	0	.02	.09	.29
1800		11.0	1.2	1.4	15.1		135.5	62.2	72.4	11.5	8011	10970	.08	.08	.14	.12	.20	.17	.13	.10	.31
1830		11.3	1.2	1.2	14.0		134.3	61.2	72.3	11.4	7920	10900	.07	.12	.14	.12	.07	.13	.02	.10	.35
1900		10.9	1.2	1.1	13.0		136.0	63.0	72.4	11.5	8000	10950	.07	.12	.14	.11	.09	.09	.02	.10	.34
1930		10.9	1.2	1.1	12.8		135.9	62.7	72.6	11.5	8036	11000	.07	.12	.14	.10	.09	.11	.02	.09	.3
2000		11.0	1.3	1.4	13.0		135.8	62.4	72.6	11.5	8029	10990	.06	.11	.15	.10	.1	.14	.02	.09	.34
2030		11.0	1.3	1.2	15		135.1	62.2	72.3	11.4	7983	10920	.11	.19	.16	.11	.14	.12	.02	.1	.3
2100		10.9	1.3	1.3	14.7		134.9	62.2	72.1	11.5	7960	10890	.05	.09	.17	.1	.1	.13	.02	.09	.3
2130		11.0	1.3	1.1	13.5		134.3	61.9	72.5	11.4	8043	11020	.09	.15	.15	.1	.09	.11	.02	.1	.29
2200		10.9	1.3	1.3	14.5		134.3	61.6	72.1	11.4	7923	10850	.06	.12	.14	.1	.09	.14	.02	.11	.36
2230		10.6	1.3	1.2	13.4		135.0	62.2	72.3	11.4	8001	10960	.06	.11	.15	.15	.1	.15	.02	.1	.35
2300		10.8	1.2	1.3	14.0		135.0	62.2	72.1	11.4	7982	10930	.06	.11	.15	.1	.1	.14	.02	.1	.36
2330		10.7	1.3	1.2	14.5		135.3	62.4	72.2	11.4	7986	10940	.06	.12	.15	.1	.1	.14	.02	.1	.36
2330	3	OFF ENDURANCE																			
2335		SHUT DOWN																			

Remarks Added 1 qt 7808 @ 2010 Hrs.
" 1½ qt " @ 2030 ".
" 1 qt " @ 2130 "
" 1 " " @ 2300 HRS

Page No

Pratt & Whitney Aircraft

U
A.

LOG OF ENGINE TEST

EXPERIMENTAL ENGINE DEPARTMENT

Sheet No. 3/4 E

Date 01-2-78

Engineer Bill Gamm

Operators Reid/Giles

D-4 Stand   Engine/Rig No. F-34024   Build 10   Project

Type of Test 50 HR ENDURANCE RUN

"A" SWITCH °F

Time	Test Pt. Mission Point No.	T_1 Oil Tank A-1	T_2 Oil Pump Dis. A-2	T_3 2/3 Comp Sup A-3	T_4 1-4-5 Oil Sup. A-4	T_{51} 2/3 Comp Air A-5	T_{52} 2/3 Comp Air A-6	T_6 1-4-5 Air ORI. A-7	T_{71} Forw Cham Temp A-8	T_{72} Forw Cham Temp A-9	T_{81} Rear Cham Temp A-10	T_{82} Rear Cham Temp A-11	T_9 Bore Temp A-12	T_{10} Forw Dome Temp A-13	T_{12} Forw Dome Temp A-14	T_{11} Rear Dome A-15	T_{13} Rear Dome A-16	T_{15} Brea. Air Orif. A-17	T_{16} 2/3 Scav. Pum Disc A-18	T_{14} Bore Air Out A-19
2330		262	[illegible]	[illegible]	[illegible]	[illegible]	267	67	231	[illegible]	[illegible]	[illegible]	[illegible]	197	19[illegible]	319	321	215	[illegible]	[illegible]
2330		-	[illegible]	[illegible]	[illegible]															
2335			[illegible]																	
1715	3		START 1/2/78																	
1730			ON ENDURANCE																	
1730		240	240	232	231	243	246	67	210	211	329	329	244	218	219	290	292	205	257	198
1800		260	260	248	247	263	265	72	217	216	359	359	300	139	137	303	304	215	270	249
1830		264	264	252	252	266	268	73	201	200	343	343	316	155	155	286	288	241	275	248
1900		250	250	238	238	254	255	71	214	213	343	344	326	174	174	286	288	208	262	250
1930		262	262	251	250	265	266	69	213	213	343	343	319	170	170	232	284	210	274	241
2000		259	258	246	245	261	262	69	216	216	350	351	328	188	[illegible]	293	295	210	270	[illegible]
2030		254	254	241	241	257	259	69	222	222	351	351	329	184	185	298	300	209	265	255
2100		258	259	244	243	[illegible]	263	69	254	255	357	357	334	187	187	333	325	[illegible]	[illegible]	260
2130		274	274	264	263	276	279	68	232	231	356	356	331	187	187	297	290	218	288	255
2200		258	258	246	245	261	263	64	230	230	340	340	328	198	198	[illegible]	294	212	269	249
2230		261	267	249	248	264	265	63	233	233	347	347	322	202	202	[illegible]	290	210	273	[illegible]
2300		265	264	249	248	267	268	63	237	237	347	348	325	205	205	290	291	213	271	248
2330	04	256	256	243	243	260	261	63	237	236	348	348	323	205	205	292	294	214	267	248
2330	03	OFF ENDURANCE																		
2335		STOP																		

Remarks

Page No.

Report

LOG OF ENGINE TEST

Pratt & Whitney Aircraft

U A.

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 414 E

Date 1-26-78

Engineer Bill Gamble

Operators Red/Giles

D-4 Stand — Engine/Rig No. F34024 Build 10 Project

Type of Test 50 HR. ENDURANCE RUN

Time A.M. P.M.	Tot. Hrs	T17 Front Dome Air Clr A-20	T18 Rear Dome Air A-21	T19 G/B OIL IN A-22	T20 G/B OIL out A-23		B-11 No. 2 BRG. B-1	B-12 No. 2 BRG. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower shaft Brg B-10	B-61 upper Tower Shaft Brg B-11	B-62 upper Tower Shaft BRG B-12		T21 G/B BRG #1 A-24	T22 G/B BRG #2 A-25
2330		193	[illegible]	[illegible]	[illegible]		[illegible]		[illegible]		[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	168	284	284		116	120
2330			OFF	[illegible]																	
2335			[illegible]																		
1745	3		START			1/27/78															
1730		ON ENDURANCE																			
1730		222	286	78	94		255		280		255	248	260	256	289	153	266	266		101	105
1800		77	259	84	104		267		291		266	261	272	268	298	163	276	277		113	114
1830		154	248	84	105		274		297		272	268	278	274	304	168	284	284		114	115
1900		160	245	84	104		262		278		253	249	258	259	287	160	264	264		113	114
1930		160	243	83	103		274		295		267	258	267	263	295	164	272	273		111	113
2000		196	251	83	104		267		289		264	259	270	266	298	163	276	276		113	113
2030		173	263	83	104		263		287		262	257	268	264	295	162	274	275		113	113
2100		239	266	84	105		267		287		263	258	269	265	296	162	275	275		114	114
2130		168	255	83	104		285		303		274	271	279	274	306	167	280	280		113	114
2200		193	245	83	103		266		288		263	259	269	265	296	164	275	275		112	112
2230		199	248	82	102		271		293		269	263	274	270	301	165	281	281		111	112
2300		199	257	82	102		268		289		264	259	269	265	297	162	275	275		110	112
2330		199	254	82	102		265		288		263	259	269	265	296	162	274	275		111	113
2330	3	OFF ENDURANCE																			
2335			Stop																		

Remarks

Page No.

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

U
A

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 14 F

Date 1/30/78

Engineer [illegible] Gaiance

Operator Corky/Rainardi

D-4 Stand   Engine/Rig No. 34024   Build 10   Project

Type of Test 50 hour Endurance Test; Compt. Lube Sys Rig

Time Actl. P.M.	Test Hrs (Mission Point No.)	P21 LAB Cooling (G2)	P10 Forward dome (G11)	P11 REAR Dome (G12)	P3 2,3 OIL (G13)	P4 1,4,5 OIL (G14)	P2 OIL Pump (G15)	P14 Bore Exit (G16)	P19 6/8 OIL IN (G17)	P20 6/8 OIL out (G18)	P5 2,3 Breather (G19)	P9 Bore Chamb (G20)	P7 Forward chamber (G21)	P8 Rear Chamb (G22)	P15 Breather orifice (G37)	P16 2,3 Scavenge (G38)	P17 Forward Dome orifice (G39)	P18 Rear Dome orifice (G40)	P1 OIL TANK (G41)	P6 4,4,5 AIR (G42)	ΔP2 ± PSI	ΔP16 ± PSI
		PSIG	PSIG		PSIA	PSIA	PSIA	PSIA	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA		
0900	2	0	0	0	14.7	14.7	14.7	14.7	0	0	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	0	±
0900	Start Rotation					1/30/78	1st Shift															
0920	3	ON	Endurance																			
0935	↓	25	13	24	53	53	136	15	15	0	8.0	28	18	29	4.0	13.8	28	48	10	80	.1	0
1000		25	4	20	51	32	136	15	15	0	6.0	25	17	25	4.0	13.5	25	41.5	10	77	.1	0
1030		25	9	20	51	32	134	16	15	0	6.0	25	18	25	3.6	13.5	24	41.5	10	77.5	.1	0
1100		25	4	20	50	32	133	16	15	0	6.0	22	16	22	3.4	13.5	18	35	10	76	.1	0
1130		25	10	20	50.5	31.5	132	15	15	0	6.0	25	17	25	3.7	13.6	24.7	41	10	76	.1	0
1200		25	9	19	50.5	32	132	15.5	15	0	6.8	25.5	17	25	3.7	13.6	23.5	41	10	78	.1	0
1230		25	10	19	51	32	132	15.5	15	0	6.8	26	18	25	3.7	13.6	26.0	40.5	10	78	.1	0
1300		25	10	22	51	32	134	15.5	15	0	8.5	26	18	26.5	4.0	13.6	26.0	45.0	10	78	.1	0
1330	↓	25	10	27	51	32	134	15.5	15	0	7.0	24.5	18	23.5	3.6	13.6	27.0	38.5	10	77.5	.1	0
[illegible]	3	OFF	Endurance																			
[illegible]	5	ON	Endurance																			
[illegible]		25	4.0	18	51	32	133	15.5	15	0	6.4	25	16	24	3.7	13.7	20	39.5	10	78	.1	0
[illegible]		25	12.0	21.0	51	32	133	15.5	15	0	6.9	26	18	26	3.8	13.8	29	43	10	78	.1	0
1500		25	12.0	20.0	50	32	132	15.5	15	0	7.6	25	19	26	4.0	13.8	29	42	10.5	78	.1	0
1530		25	10.0	20.0	50	32	132	15.5	15	0	8.8	25	18	25	3.8	13.8	28	41	10.5	78	.1	0
1600		25	10.0	20.0	50	32	132	15.5	15	0	7.0	26	18	26	4.0	13.8	27	41.5	10.5	78	.5	0
1630		25	10.0	20.0	51	32	132	15.5	15	0	6.8	26	18	25	4.5	13.7	27	42	10.0	78	.5	0
1700		25	11.0	21.0	51	32	132	15.5	15	0	6.8	25	18	25	3.8	13.7	27	42	9.75	77.5	.5	0
1730		25	11.0	21.0	51	32	132	15.5	15	0	6.8	24.5	18	25	3.9	13.7	26.5	42	9.75	78	.5	0
1800	↓	25	11.0	21.0	51	32	132	15.5	15	0	6.8	25	18	25	3.9	13.7	26.5	42.5	9.75	78	.5	0

Remarks Added 2 Gal 7808 @ 1100   Added 1 gal @ 1715
" 3 Gal " @ 1230   Added ½ gal @ 1815
" 4 Gal " @ [illegible]

CT 3 Endur 1335 / 0930 / 4.05   RT 1335 / 0900 / 4.35

PT 5 Endur 1800 / 1335 / 4.25   R.T 1800 / 1335 / 4.25

Page No.   Report No.

PWA FRDC 9/A (REV 3/7)

Pratt & Whit[illegible] Aircraft

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LOG OF E[illegible]GINE TEST
EXPERIMENTAL [illegible]ST DEPARTMENT

Sheet No. 2/4F

Date 1/30/[illegible]8

Engineer Bill Gamble

Operators Conley/Bernard

D-4 Stand   Engine/Rig No. F-34024   Build 10   Project

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M./P.M.	Tot. Hrs / Mission Point No.	"U" Tubes Dp 6 (Inches HG) U-63	Dp 15 U-64	Dp 17 U-65	Dp 18 U-66		Flow Meters F1 Oil Supply PPM	F2 1,4,5 Oil PPM	F3 2,3 Oil PPM	F4 G/B PPM	Pump Speed RPM	Rig Speed RPM	Vibrations (Mils) Rig No. 3 Brg 350° V-1	Rig No. 3 Brg 260° V-2	Rig No. 2 Brg Vert V-3	Rig Front Vert V-4	Rig Front Horz V-5	Rig Rear Vert U-6	Rig Rear Horz V-7	G/B Vert U-8	G/B Horz -9
0900	Z	0	0	0	0		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0900		Start	Rotation			1/30/78	1st shift														
09[illegible]	3	ON	Endurance																		
0930		10.8	1.3	1.5	15.3		137.8	64.1	73.2	11.1	8080	11060	.06	.1	.16	.11	.09	.15	.12	.12	.32
1000		10.3	1.2	.7	13.0		136	63.3	71.9	11.5	7990	10940	.15	.17	.22	.13	.29	.17	.22	.11	.35
1030		10.1	1.2	1.1	13.6		136.7	63.5	72.5	11.6	8077	11050	.09	.09	.15	.12	.2	.17	.13	.07	.29
1100		10.0	1.2	1.2	11.4		136.3	63.6	72.2	11.7	8109	11130	.12	.15	.2	.7	.26	.15	.19	.07	.32
1130		10.0	1.2	1.3	13.7		134.4	62.3	71.6	11.7	7952	10910	.1	.18	.15	.11	.11	.12	.2[illegible]	.1	3
1200		11.0	1.2	1.3	13.6		134.7	62	72.0	11.7	7913	10910	.7	.1	.15	.1	.2	.17	.15	.1	.3[illegible]
1230		10.5	1.2	1.5	13.0		134.9	62.3	72.1	11.8	8003	10990	.7	.15	.12	.1	.7	.15	.21	.07	.31
1300		10.5	1.2	1.3	15.2		136.2	63.0	72.4	11.8	7994	11[illegible]	.6	.15	.12	.1	.7	.17	.25	.12	.37
1330		12.3	1.2	1.3	12.8		136.2	63.3	72.5	11.9	8[illegible]	11060	.7	.1	.15	.1	.09	.12	.2	.07	.3[illegible]
13[illegible]	3	OFF	Endurance																		
1[illegible]	5	ON	Endurance																		
1400		11.1	1.2	.8	12.8		134.6	61.9	72.0	11.8	7960	10930	1.5	1.7	2.2	.1	3.2	.15	.2	.07	.27
1430		11.0	1.2	2.0	14.7		134.8	62.1	72.0	11.8	7974	10950	.06	.1	.15	.1	.1	.1[illegible]	.15	.1	.37
1500		11.0	1.2	2.0	14.4		134.7	62.0	72.1	11.9	7939	10890	.05	.1	.15	.1	.1	.17	.15	.1	.37
1530		11.0	1.2	2.0	14.0		134.7	61.7	72.2	11.9	7955	10890	.05	.12	.15	.1	.1	.16	.2	.1	.4
1600		11.0	1.1	1.8	14.3		134.6	62.1	72.0	11.9	7997	1094.0	.05	.15	.12	.1	.1	.15	.22	.12	.34
1630		11.0	1.2	1.8	13.8		134.8	62.0	71.9	11.9	7989	1098.0	.05	.15	.12	.1	.1	.15	.22	.1	.35
1700		10.6	1.1	1.8	14.0		134.2	61.7	71.8	11.9	7995	1095.0	.05	.15	.12	.1	.1	.15	.22	.1	.35
1730		11.0	1.2	1.7	13.8		135.0	62.2	72.2	11.9	7995	1095.0	.05	.15	.12	.1	.1	.15	.22	.11	.35
1800		11.0	1.2	1.8	14.0		134.6	62.0	72.0	11.9	7975	1093.0	.05	.15	.12	.1	.1	.15	.22	.11	.35

Remarks

Page No.

Report No.

PWA [illegible] REV 3/[illegible]

FD 4[illegible]

AD-A060 172 PRATT AND WHITNEY AIRCRAFT GROUP WEST PALM BEACH FL G--ETC F/G 11/8
COMPARTMENTAL LUBRICATION SYSTEM.(U)
JUN 78 E M BEVERLY F33615-75-C-2075
UNCLASSIFIED PWA-FR-9555 AFAPL-TR-78-32 NL

4 OF 4
AD
A060 172

END
DATE
FILMED
12-78
DDC

Pratt & Whitney Aircraft U.A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 3/4 F
Date 01/ 3/78
Engineer Bill Gamble
Operators Cawley/Bergland

Stand D-4 Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN

"A" Switch °F

Time A.M./P.M.	Test Nos. (Point No. / Mission)	T1 Oil Tank	T2 Oil Pump Dis.	T3 2/3 Comp Sup	T4 1-4-5 Oil Sup.	TS1 2/3 Comp Air	TS2 2/3 Comp Air	T6 1-4-5 Air ORI.	TT1 Forw Cham Temp	TT2 Forw Cham Temp	T81 Rear Cham Temp	T82 Rear Cham Temp	T9 Bore Temp	T10 Forw Dome Temp	T12 Forw Dome Temp	T11 Rear Dome	T13 Rear Dome	T15 Brea. Air ORF.	T16 2/3 Scav. Rim Disc	T14 Bore Air Out
		A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9	A-10	A-11	A-12	A-13	A-14	A-15	A-16	A-17	A-18	A-19
0900	2	45	42	42	43	46	47	43	46	46	46	45	41	46	45	44	44	43	46	44
0900		START ROTATION																		
0930	3	ON ENDURANCE																		
0935		249	249	239	238	253	255	52	246	246	348	349	248	247	246	305	307	202	263	211
1000		252	252	245	248	255	256	57	214	214	318	319	288	150	152	248	250	202	257	224
1030		260	260	249	247	263	265	61	230	229	364	364	305	189	190	313	315	206	270	248
1100		261	260	247	248	263	264	63	220	219	344	344	310	155	157	279	281	205	268	235
1130		262	262	250	249	267	268	63	267	268	246	347	219	248	248	289	291	222	275	[illegible]
1200		259	259	246	244	263	264	65	254	254	346	347	347	230	230	291	293	209	269	257
1230		244	264	253	252	267	269	67	247	246	325	326	332	249	248	330	331	208	278	269
1300		256	257	248	246	262	264	68	220	219	350	350	336	186	186	295	297	230	272	[illegible]
1330		263	262	257	250	266	268	71	229	229	356	356	333	194	196	300	352	227	272	259
1335	3	OFF ENDURANCE																		
1335	5	ON ENDURANCE																		
1400		259	259	247	248	263	265	71	212	211	337	338	325	130	131	272	273	209	269	248
1430		256	261	249	248	263	266	70	238	238	343	343	334	210	211	285	226	208	272	256
1500		251	252	246	244	259	251	71	239	[illegible]	[illegible]	[illegible]	329	212	212	251	283	234	271	[illegible]
1530		262	262	249	248	267	268	72	[illegible]	[illegible]	240	310	[illegible]	[illegible]	207	279	281	[illegible]	275	[illegible]
1600		262	262	247	247	265	266	73	233	233	340	340	329	200	200	284	285	220	271	255
1630		264	264	252	251	266	268	23	233	233	341	342	336	198	199	281	283	214	275	258
1700		262	262	250	249	265	267	72	232	232	345	345	334	198	198	285	286	213	274	257
1730		262	262	250	249	265	267	70	231	230	339	339	331	196	196	277	279	213	273	251
1800		262	264	253	252	268	269	68	230	230	338	338	336	194	195	278	279	214	278	257

Remarks

Page No. Report No

Pratt & Whitney Aircraft

U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 4/4 F

Date 01, 30/78

Engineer BILL GAMBLE

Operators COLEY/BERNARD

D-4 Stand   ~~Engine~~/Rig No. F34024   Build 10   Project

Type of Test 50 HR ENDURANCE RUN

Time A.M./P.M.	Tot. Hrs (MISSION POINT NO)	T17 Frw Dome Air OEW A20	T18 Rear Dome Air A-21	T19 G/B OIL IN A-22	T20 G/B OIL OUT A-23	B-11 No.2 BRG. B-1	B-12 No.2 BRG. B-2	B-21 No.3 Brg. B-3	B-22 No.3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft BRG B-12	T21 G/B BRG #1 A24	T22 G/B BRG #2 A25
		"A" SW OF				"B" SWITCH OF													
0900	2	42	42	43	42	45		45		45	44	45	44	45	44	45	46	41	41
0900			START ROTATION																
0930	3		ON ENDURANCE																
0935		279	289	72	89	260		285		252	265	260	260	292	141	270	271	98	103
1000		146	207	76	96	256		281		250	260	256	265	288	156	266	266	105	107
1030		174	280	78	99	267		290		265	260	271	267	298	156	274	275	109	110
1100		98	224	79	99	262		286		261	261	257	266	269	161	271	272	109	110
1130		244	252	79	100	270		293		269	264	274	276	305	163	280	280	111	112
1200		224	253	80	101	265		287		263	258	269	265	296	162	275	275	112	112
1230		288	301	81	103	273		296		271	266	277	273	301	163	281	281	115	114
1300		176	261	81	103	266		290		264	258	271	266	297	163	277	277	113	113
1330		188	261	82	104	269		195		270	265	276	272	303	166	282	285	114	115
1335	3	OFF ENDURANCE																	
1335	5	ON	ENDURANCE																
1400		89	221	82	103	265		290		265	260	270	266	297	167	276	276	113	114
1430		209	245	82	104	269		291		266	261	271	267	297	165	277	277	115	115
1500		209	242	82	104	268		291		266	261	274	265	298	167	278	[illegible]	115	115
1530		199	238	83	104	269		291		266	261	272	268	298	167	277	278	114	115
1600		192	250	83	104	268		290		266	261	272	267	297	167	276	277	114	116
1630		191	240	83	105	273		295		271	266	276	272	302	170	281	282	116	115
1700		190	242	83	105	271		294		269	264	275	271	301	169	281	281	116	115
1730		188	236	82	104	270		293		268	263	273	269	300	169	279	280	114	114
1800		187	238	82	104	275		298		273	268	278	274	304	170	283	283	114	114

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Remarks

Report No.

PWA FH (REV 3/71)   FD 4897

Pratt & Whitney Aircraft — U.A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 74 G
Date 1/30/78
Engineer B. Gaining
Operators Rusler

D-4 Stand — Engine/Rig No. 34024 — Build 10 — Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time A.M./P.M.	Test Hrs (Missing Point No.)	P_{21} LAB Cooling	P_{10} Forward dome	P_{11} REAR Dome	P_3 2,3 OIL	P_4 1,4,5 OIL	P_2 OIL Pump	P_{14} Bore Exit	P_{19} G/B OIL IN	P_{20} G/B OIL out	P_5 2,3 Breather	P_9 Bore Chamb	P_7 Forward chamber	P_8 Rear Chamb	P_{15} Breather orifice	P_{16} 2,3 Scavenge	P_{17} Forward Dome orifice	P_{18} Rear dome orifice	P_1 OIL Tank	P_6 1,4,5 AIR	ΔP_2 ± psi	ΔP_{16} ± psi
		PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIA	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA		
		G2	G11	G12	G13	G14	G15	G16	G17	G18	G19	G20	G21	G22	G37	G38	G39	G40	G41	G42		
1830	5	25	11	21	51	32	132	16	15	0	6.8	25	18	25	3.8	13.7	27	42	9.25	78	.5	0
1900		25	11	21	51	32	132	16	15	0	6.8	25	18	25	3.8	13.6	27	42.5	9.75	78	.5	0
1930		30	11	21	51	32	132.5	16	15	0	6.8	25	18	25	3.8	13.6	26.5	42.5	9.8	78	.5	0
2000		30	11	21	51	32	132.5	16	15	0	6.8	25	18	25	3.8	13.6	27	42.5	9.8	78	.5	0
2030		30	11	21	51	32	132	16	15	0	6.8	25	18	25	3.8	13.6	26	44	10	79.5	.5	0
2100		30	11	20	52	32	134	16	15	0	6.8	25.5	18	24.8	3.8	13.4	27	40	10	78	.5	0
2130		30	11	20	52	32	134	16	15	0	6.8	25	18	25	3.8	13.4	26	41	10	78	.5	0
2200		30	11	20	52	32	131	16	15	0	6.8	25	18	25	3.8	13.6	26	44	10	79	.5	0
2230		30	10	20	52	32	133	16	15	0	6.6	25	18	25	3.7	13.6	26.5	42	10	78.5	.5	0
2300		30	11	21	51	32	132	16	15	0	6.8	25	18	25	3.8	13.6	26.5	42.5	9.5	78	.5	0
2330		30	11	20	51	32	133	16	15	0	6.8	25	18	25	3.8	13.6	26.5	41	9.5	78	.5	0
2335	5	OFF		ENDURANCE																		
2338			SHUT	DOWN																		

Remarks: POINT #5 END 2335 1800 5:35 R.T. 2338 1800 5:38

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PWA FRDC 97A REV 3 71

Pratt & Whitney Aircraft U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 2/46

Date 1/30/78

Engineer Bill Gaines

Operators Rasler

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN COMP LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs (Endurance Point No.)	Dp 16 (U-?)	"U" Tubes Dp 15 (INCHES HG) U-64	Dp 17 U-65	Dp 18 U-66	Flowmeters F1 Oil Supply PPM	F2 1,4,5 Oil PPM	F3 2,3 Oil PPM	F4 G/B PPM	Rump Speed RPM	Rig Speed RPM	Vibrations (MILS) Rig No. 3 Brg 350° V-1	Rig No. 3 Brg 260° V-2	Rig No. 2 Brg Vert V-3	Rig Front Vert V-4	Rig Front Horz V-5	Rig Rear Vert V-6	Rig Rear Horz V-7	G/B Vert V-8	G/B Horz V-9
1830	5	10.9	1.2	1.9	13.9	134.8	62.0	72.1	11.9	7991	10960	.05	.15	.12	.1	.1	.15	.25	.12	.39
1900		10.8	1.2	1.8	13.8	134.5	61.8	72.0	11.8	7989	10950	.05	.15	.12	.1	.1	.15	.25	.12	.35
1930		10.9	1.2	1.8	13.9	134.5	61.7	72.1	11.8	7996	10920	.06	.15	.14	.10	.10	.15	.02	.12	.37
2000		11.0	1.2	1.8	14.2	134.7	62.0	71.9	11.8	7976	10930	.05	.15	.12	.1	.1	.15	.02	.12	.38
2030		11.5	1.3	1.7	15.5	133.9	60.9	73.1	11.8	7982	10950	.09	.17	.14	.1	.05	.13	.02	.13	.34
2100		11.0	1.4	1.8	12.8	135.6	62.5	72.3	11.8	8016	11050	.08	.13	.14	.09	.09	.12	.24	.12	.28
2130		11.2	1.3	1.8	13.8	134.9	62.1	72.2	11.8	8011	11030	.07	.15	.14	.1	.09	.14	.25	.12	.34
2200		11.5	1.3	1.8	15.0	133.6	61.0	72.0	11.7	7972	10930	.07	.16	.14	.1	.08	.14	.26	.13	.36
2230		10.5	1.2	1.8	14.0	134.6	62.0	71.9	11.8	7993	10980	.05	.16	.14	.1	.08	.15	.02	.1	.38
2300		11.0	1.2	1.8	14.7	133.9	61.6	71.7	11.8	7952	10910	.05	.15	.14	.1	.08	.15	.02	.12	.4
2330		11.0	1.2	1.8	13.5	134.4	61.7	71.9	11.8	7985	10960	.05	.12	.12	.1	.08	.15	.23	.12	.35
2335	5	OFF ENDURANCE																		
2338		SHUT DOWN																		

Remarks: Added 1 gal oil to Rig tank 1945 Added ½ gal @ 2230 hrs
Added 1 gal oil @ 2015
Added ½ gal @ 2130

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PWA FORM REV 3/71 FD

Pratt & Whitney Aircraft U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 3/5 G

Date 01-30-78

Engineer Big Gamble

Operators Giles

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Type of Test 50 HR ENDURANCE RUN

"A" Switch OF

Time Adj. FM.	Tot. Hrs	T_1 Oil Tank A-1	T_2 Oil Pump Dis. A-2	T_3 2/3 Comp Sep A-3	T_4 1-4-5 Oil Sep. A-4	T_{51} 2/3 Comp Air A-5	T_{52} 2/3 Comp Air A-6	T_6 1-4-5 Air ORI. A-7	T_{71} Forw Cham Temp A-8	T_{72} Forw Cham Temp A-9	T_{81} Rear Cham Temp A-10	T_{82} Rear Cham Temp A-11	T_9 Bore Temp A-12	T_{10} Forw Dome Temp A-13	T_{12} Forw Dome Temp A-14	T_{11} Rear Dome A-15	T_{13} Rear Dome A-16	T_{15} Brea. Air ORF. A-17	T_{16} 2/3 Scav. Pum Disc A-18	T_{14} Bore Air Out A-19
1830	5	260	261	249	248	265	266	66	229	229	338	338	340	193	194	276	278	216	272	258
1900		262	263	251	251	267	269	65	229	228	338	338	339	192	193	276	277	214	277	259
1930		259	260	248	248	262	263	65	222	226	336	[illegible]	336	191	191	275	277	214	273	253
2000		258	258	244	246	261	263	64	227	226	336	336	336	191	191	276	277	212	270	251
2030		267	267	255	254	271	271	68	241	241	363	364	334	201	202	314	316	222	[illegible]	260
2100		253	259	248	248	262	264	65	230	230	343	344	343	197	197	281	283	213	271	263
2130		245	259	246	246	262	263	64	231	231	341	341	330	196	197	281	283	213	272	257
2200		266	266	255	255	270	272	64	227	229	346	346	334	191	192	288	289	218	282	252
2230		259	259	246	245	261	263	63	218	216	339	339	320	192	195	276	277	212	271	242
2300		258	259	246	245	261	263	60	216	216	344	345	324	180	180	288	299	216	271	246
2330	↓	258	259	246	246	261	263	63	225	225	346	346	324	189	189	289	290	217	273	248
2335	5	OFF ENDURANCE																		
2338		SHUTDOWN																		

Page No. Remarks Report No.

Pratt & Whit[illegible] Aircraft

U
A

LOG OF E[illegible]GINE TEST

EXPERIMENTAL [illegible]EST DEPARTMENT

Sheet No. 4/[illegible]

Date 01-30-78

Engineer Bill Gamble

Operators Giles

D-4 Stand   Engine/Rig No. E34024   Build 10   Project

Type of Test 50 HR. ENDURANCE RUN

Time A.M. P.M.	Tot. Hrs.	T17 Front Dome Air °F A-20	T18 Rear Dome Air A-21	T19 G/B Oil In A-22	T20 G/B Oil Out A-23	B-11 No. 2 BRG. B-1	B-12 No. 2 BRG. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft BRG B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft BRG B-12	T21 G/B BRG #1 A24	T22 G/B BRG #2 A25
1830	5	186	234	82	104	250		250		267	262	272	268	298	167	278	278	114	114
1900		184	234	82	104	274		297		272	267	277	274	303	169	284	285	114	115
1930		183	235	81	103	269		292		267	262	273	269	299	166	279	279	111	111
2000		183	234	81	102	265		287		202	257	267	264	294	163	274	271	111	113
2030		181	288	81	102	276		299		274	220	281	276	305	166	285	285	111	111
2100		191	234	81	103	267		290		264	259	270	266	297	165	276	276	112	114
2130		189	241	80	100	267		290		264	259	270	266	296	164	276	276	108	110
2200		183	250	80	101	275		299		274	269	280	276	305	166	286	286	109	110
2230		137	228	79	100	266		289		263	258	268	265	296	163	274	275	107	109
2300		176	252	79	99	266		288		263	258	269	265	295	161	274	275	107	108
2330		182	249	78	99	267		289		265	260	270	266	297	162	276	276	107	108
2335	5	OFF ENDURANCE																	
2338		SHUT DOWN																	

Remarks

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PWA FR[illegible] REV 3 71   FD 4P[illegible]

Pratt & Whitney Aircraft
FLORIDA RESEARCH AND DEVELOPMENT CENTER

U
A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 1/4 H
Date 1/31/78
Engineer B. Gaines
Operators Canby / Renard

D-4 Stand ~~Engine~~/Rig No. 34024 Build 10 Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time	Test Hrs / Mission Point No.	P_{21} LAB Cooling	P_{10} Forward dome	P_{11} REAR Dome	P_3 2,3 OIL	P_4 1,4,5 OIL	P_2 OIL Pump	P_{14} Bore Exit	P_{19} 6/8 OIL IN	P_{20} 6/8 OIL out	P_5 2,3 Breather	P_9 Bore Chamb	P_7 Forward Chamber	P_8 Rear Chamb	P_{15} Breather orifice	P_{16} 2,3 Breather	P_{17} Forward Dome orifice	P_{18} Rear Dome orifice	P_1 OIL TANK	P_6 4,5 AIR	ΔP_2	ΔP_{16}
		PSIG	PSIG		PSIA	PSIA	PSIA	PSIA	PSIG	PSIG	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA	PSIA		
		G2	G11	G12	G13	G14	G15	G16	G17	G18	G19	G20	G21	G22	G37	G38	G39	G40	G41	G42		
0845	2	0	0	0	14.7	14.7	14.7	14.7	0	0	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	14.7	0	0
0900	5	Start Endurance				1/31/78 is start																
1000		ON Endurance																				
1005		30	12	18	51	32	135	15	15	0	6.2	24	18	24	3.5	13.5	28	39	9.5	77	.1	0
1035		30	10	23	51	33	131	15	15	0	7.0	26.5	18	28	5.0	13.5	31	48	9.7	78	.1	0
1105		30	12	15	51	32	133	15	15	0	6.8	23	18	22	3.6	13.8	27	35	9.5	76	.1	0
1135		30	11	18	51	32	135	15	15	0	6.8	25	17.5	24	3.7	13.6	24	40	9.5	77	.1	0
1205		30	12	18	51	32	134	15	15	0	6.6	24.5	17.5	24	3.8	13.8	27.5	39.7	9.5	77	.1	0
1235		30	10	17	51	32	135	15	15	0	6.6	23	17.5	23	3.7	13.8	25	38	9.5	77	.1	0
1305		30	13	18	51	32	134	15	15	0	6.8	24	19	24	3.8	13.7	29	39.5	9.5	77	.1	0
1335		31	8	18	51	32	134	15	15	0	6.6	26	17.5	24.5	3.8	13.8	33	39.5	9.5	77	.1	0
1355		30	12	20	50.5	32	133	15	15	0	6.8	25	18	25	4.0	13.8	27.5	41.5	9.5	77	.1	0
1410	5	OFF Endurance																				
1415		Shut Down																				

Remarks 0900 Countdown added 1.5 Gal oil To Rig
1030 added 2 Gal To Rig
1200 " 2 Gal " "
1400 " 5 qts To Rig

Hr = 5 E.T 1405 − 0900 = 5:05 ... 1400 − 12.00 = 4:00

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Report No.

FRDC 9/5 REV 1/71

Pratt & Whitney Aircraft

U
A.

LOG OF ENGINE TEST
EXPERIMENTAL TEST DEPARTMENT

Sheet No. 3 of 4

Date 1/31/78

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Engineer Bill Gaunce

Operators Cooley/Rinaldi

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs MISSION NO. POINT NO.	"U" Tubes D_p 6	D_p 15	D_p 17	D_p 18		FLOW METERS F_1 OIL Supply	F_2 1,4,5 OIL	F_3 2,3 OIL	F_4 G/B	Pump Speed	Rig Speed	VIBRATIONS R/C NO. 3 BRG 350°	R/C NO. 3 BRG 260°	R/C NO. 2 BRG VERT	R/C FRONT VERT	R/C FRONT HORZ	R/C REAR VERT	R/C REAR HORZ	G/B VERT	G/B HORZ
		INCHES HG					PPM	PPM	PPM	PPM	RPM	RPM	MILS								
		U 63	U-64	U-65	U-66								V-1	V-2	V-3	V-4	V-5	V-6	V-7	V-8	-9
0915	2	0	0	0	0		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0930	5	Start Rotation				1/31/78	1st Start														
1000		ON Endurance																			
1005		11.0	1.2	2.0	12.9		135.2	62.3	72.2	11.5	8026	10990	.05	.1	.15	.12	.09	.16	.14	.13	.32
1035		11.3	1.2	1.2	17.3		132.5	61.0	71.8	11.8	7981	10930	.1	.2	.15	.13	.19	.15	.25	.1	.3
1105		11.1	1.2	1.8	10.8		137.0	63.5	72.8	11.8	8136	11149	.05	.1	.15	.1	.09	.15	.15	.09	.25
1135		10.8	1.3	1.3	13.5		135.1	62.8	72.2	11.9	8068	11050	.07	.07	.15	.12	.19	.15	.13	.07	.27
1205		10.9	1.2	1.8	13.2		135.3	62.4	72.2	11.9	8015	11010	.05	.11	.13	.1	.1	.15	.19	.09	.35
1235		10.8	1.2	1.6	12.7		135.6	62.7	72.2	12.0	8035	11060	.09	.13	.13	.1	.1	.1	.2	.09	.27
1305		11.0	1.3	2.2	13.0		135.0	62.3	72.0	12.0	8019	11000	.05	.1	.15	.1	.11	.15	.15	.1	.37
1335		11.0	1.3	1.2	12.6		134.9	62.1	72.0	12.0	8048	11060	.07	.09	.15	.1	.19	.15	.14	.07	.27
1355		11.5	1.3	2.0	13.5		133.5	61.2	71.6	12.0	7994	10962	.05	.15	.13	.1	.1	.15	.2	.11	.37
1400	5	OFF Endurance																			
1405		Shut Down																			

Remarks

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Report No.

PWA FRM 776 REV 3.71

FD 48955

Pratt & Whi... Aircraft — U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. ... H

Date 01/21/78

Engineer Big Gamble

Operators Conley

D-4 Stand Engine/Rig No. F-34024 Build '10 Project

Type of Test 50 HR ENDURANCE RUN

Time A.M. P.M.	Tot. Hrs. / Mission Point No.	T_1 Oil Tank A1	T_2 Oil Pump Dis. A-2	T_3 2/3 Comp Sup A-3	T_4 1-4-5 Oil Sup. A-4	T_{S1} 2/3 Comp Air A-5	T_{S2} 2/3 Comp Air A-6	T_6 1-4-5 Air ORI. A-7	T_{7_1} Forw Cham Temp A-8	T_{7_2} Forw Cham Temp A-9	T_{8_1} Rear Cham Temp A-10	T_{8_2} Rear Cham Temp A-11	T_9 Bore Temp A-12	T_{10} Forw Dome Temp A-13	T_{12} Forw Dome Temp A-14	T_{11} Rear Dome A-15	T_{13} Rear Dome A-16	T_{15} Brea. Air ORIF. A-17	T_{16} 2/3 Scaw. Rim Disc A-18	T_{14} Bore Air Out A-19
		"A" Switch of																		
0915	2	88	60	54	59	110	110	54	106	104	99	99	52	106	102	94	84	53	106	56
0950		START		ROTATION																
1000	5	ON	END.																	
1005		258	258	247	245	261	262	66	208	207	360	361	288	169	170	310	312	205	269	235
1035		261	261	248	247	265	267	69	225	224	362	362	310	148	146	312	314	215	275	250
1105		259	259	249	247	262	262	70	224	223	353	353	322	189	190	291	309	232	268	242
1135		261	261	248	247	264	265	73	216	218	357	359	335	172	172	303	305	219	272	259
1205		262	262	250	249	265	265	75	227	227	356	357	331	195	195	301	303	216	273	253
1235		259	261	248	248	265	265	76	231	231	354	354	327	194	194	299	301	230	273	247
1305		261	261	248	248	263	263	79	207	207	354	354	338	171	171	295	297	223	274	259
1335		262	262	249	249	264	266	80	215	213	351	351	344	138	136	292	294	219	274	265
1355		258	261	248	248	264	264	80	231	231	344	345	340	199	199	286	287	217	273	259
1400	5		OFF	Endurance																
1405			Shut	Down																

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Remarks

Report No.

PWA FRM ... REV 3.71

Whit[illegible] Aircraft U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. [illegible]

Date 01/31/78

Engineer BILL GAMBLE

Operators COMPY

D-4 Stand Engine/Rig No. E34024 Build 10 Project

Type of Test 50 HR. ENDURANCE RUN

Time A.M. P.M.	Tot. Hrs	"A" SW OF T17 Forw Dome Air Ovr A-20	T18 Rear Dome Air A-21	T19 G/B Oil In A-22	T20 G/B Oil Out A-23	"B" SWITCH OF B-11 No.2 BRG. B-1	B-12 No.2 BRG. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 BRG B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft Brg B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft Brg B-12	T21 G/B Brg #1 A-24	T22 G/B Brg #2 A-25
0915	2	52	53	54	62	109		112		106	106	103	103	103	75	103	109	61	60
0930		START ROTATION																	
1000	5	ON END																	
1005		143	275	76	95	265		287		262	257	268	264	296	156	273	273	104	107
1035		74	279	80	100	268		293		268	263	275	270	299	161	279	278	109	110
1105		187	251	80	100	264		284		260	255	265	262	293	161	270	271	108	110
1135		158	263	81	102	267		291		265	260	271	269	299	165	277	277	112	112
1205		194	262	81	102	269		291		266	261	272	268	299	166	277	278	111	112
1235		180	259	82	102	269		291		267	262	272	269	300	168	277	275	111	113
1305		171	257	84	105	269		290		266	261	271	267	299	169	276	277	114	114
1335		82	252	84	106	269		292		267	262	272	269	299	170	277	278	116	115
1355		192	242	84	106	268		291		266	261	271	268	298	170	277	277	115	114
1400	5	off Endurance																	
1405		Shut Down																	

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Remarks

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PWA FM [illegible] REV 3/71

FD 4[illegible]

Pratt & Whitney Aircraft — U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 14

Date 2/10/78

Engineer B. Gaines

Operators Canby/Bonnie

D-4 Stand   Engine/Rig No. 34024   Build 10   Project

Type of Test 50 hour Endurance Test, Compt. Lube Sys Rig

Time (Act. P.S.T.)	Tot. Hrs (Mission Point No.)	P21 LAB Cooling G2	P10 Forward dome G11	P11 REAR Dome G12	P3 2,3 OIL G13	P4 1,4,5 OIL G14	P2 OIL Pump G15	P14 Bore Exit G16	P19 6/8 OIL IN G17	P20 6/8 OIL out G18	P5 2,3 Breather G19	P9 Bore Chamb G20	P7 Forward Chamber G21	P8 Rear Chamb G22	P15 Breather orifice G37	P16 2,3	P17 Forward Dome orifice G39	P18 Rear Dome orifice G40	P1 OIL TANK G41	P6 1,4,5 AIR G42	ΔP2	ΔP16
		PSIG			PSIA				PSIG						PSIA						±	± PSI
0915		0	0	0	14.7	14.7	14.7	14.7	0	0	14.7	14.7	14.5	14.7	14.7	14.7	14.7	14.7	14.7	14.7		
0930	4	START		ROTATION				2-10-78		–	1st	SHIFT										
1015		30	25	20	69	50	178	18	15	0	15	33.5	23	26.5	11	20	39	40	15	134	.2	0
1035	4	Start Endurance																				
1040		50	28	28	78	55	205	20	15	0	13	46	25.5	30.5	7	21.6	48	48	13	145	.2	0
1100		50	26	17	76	55	192	20	15	0	15	46	26	24	6.6	22.5	45	33.5	14	145	.2	0
1115		50	25	11	77.5	55	199	19	16	0	12.5	44.5	26	17	6	22.3	44	20	14.5	145	.2	0
1130		50	24	10	77	55	197	19	16	0	12.0	44	26	17	6	22.5	43	20	12.5	145	.2	0
1145		51	23	8	78	55	202	19	16	0	12.0	44	25	18	6	22.5	42.5	23	12.5	145	.2	0
1200		51	22	8	79	56	203	19	16	0	11.7	43	25	19	6	22.5	42	23	12.5	145	.2	0
1215		51	22	8	78	56	200	19	16	0	11.1	43	25	19	6	22.	41	23	12.	145	.2	0
1230		66	22	10	78	56	196	18	16	0	11.6	39	25	20	6.1	22.1	41.5	25	12.5	145	.2	0
1245		66	22	6	77	56	200	18	16	0	11.1	36	25	18	6.1	22.1	41	22	12.	145	.2	0
1300		66	22	3	78	56	204	18	16	0	11.0	28	24	16	6	22.5	40	19	12.	145	.2	0
1310	4	OFF		ENDURANCE																		
1315	4	ON		ENDURANCE																		
1330		66	34	12	82	61	205	18	16	0	13.8	37	30	21	6.2	24.5	54	27.5	14.0	165	.2	0
1345		66	28	12	83	61	211	18	16	0	13.5	37	28	21	7.2	25.8	46	29	14.0	165	.5	0
1400		66	27	12	83	61	211	18	16	0	13.5	37	28	21	7.2	25.7	46	29	14.0	165	.5	0
1415		66	27	12	83	61	210	18	16	0	13.2	37	28	21	7.2	25.8	47	29	14.0	165	.5	0
1430		66	27	13	82	61	209	18	16	0	13.3	39	28	21	7.2	25.8	47	29	14.0	165	.5	0
1445		66	27	13	83	61	213	18	16	0	13.2	40	27	21.5	7.2	25.8	45.5	30.5	14.0	165	.5	0
1446		OFF		ENDURANCE																		
1500																						

Remarks STOP   TRT 1500 / 0930 / 530

END M# 4 1310 / 1035   M# 2 1446 / 1315   1.55

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FWA FRDC 97A REV 3 71   FD 48919

Pratt & Whitney Aircraft U A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 2/3

Date 2/10/78

D-4 Stand Engine/Rig No. F-34024 Build 10 Project

Engineer Bill Gamble

Operators Conley/Brown

Type of Test 50 HR ENDURANCE RUN COMPT LUBE SYS RIG

Time A.M. P.M.	Tot. Hrs MISSION POINT NO.	"U" Tubes D_p 6 U-63	D_p 15 INCHES U-64	D_p 17 HG U-65	D_p 18 U-66		Flow Meters F_1 OIL Supply PPM	F_2 1,4,5 OIL PPM	F_3 2,3 OIL PPM	F_4 G/B PPM	Pump Speed RPM	Rig Speed RPM	Vibrations (MILS) RIG NO. 3 BRG 350° V-1	RIG NO. 3 BRG 260° V-2	RIG NO. 2 BRG VERT V-3	RIG FRONT VERT V-4	RIG FRONT HORZ V-5	RIG REAR VERT V-6	RIG REAR HORZ V-7	G/B VERT V-8	G/B HORZ V-9
0910	2	0	0	0	0		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
0930	4	Start Rotation 2/10/78					1st shift														
1015		46.7	.8	3.3	9.5		144.	59.9	83.6	11.5	8548	11700	.07	.19	.2	.12	.15	.27	.3	.17	.37
1035		Start Endurance																			
1040		48.5	2.0	4.8	7.7		156.8	66.8	88.7	11.9	9436	12920	.1	.1	.19	.1	.12	.27	.45	.3	.25
1100		50.0	2.0	4.8	3.0		152.1	63.6	86.8	12.0	9090	12460	.3	.22	.22	.17	.15	.37	.55	.25	.2
1115		50.5	1.8	4.8	2.0		155	64.9	88.4	12.1	9474	12990	.12	.12	.22	.1	.12	.26	.45	.37	.25
1130		48.4	1.8	4.8	0		154.4	64.7	88.6	12.2	9540	13060	.11	.11	.21	.1	.12	.25	.41	.45	.32
1145		49.0	1.7	4.6	.4		155.3	65.6	86.7	12.2	9496	13000	.07	.12	.22	.09	.13	.23	.5	.33	.31
1200		49.0	1.8	4.5	.3		155.7	66.0	88.3	12.2	9470	12970	.09	.12	.21	.09	.12	.23	.5	.37	.29
1215		48.5	1.6	4.4	.3		154.2	65.1	88.1	12.2	9477	12980	.1	.15	.21	.09	.13	.27	.45	.17	.3
1230		49.7	1.7	4.4	1.2		152.8	63.6	88.0	12.2	9430	12910	.1	.07	.21	.1	.12	.3	.39	.16	.32
1245		49.4	1.8	4.4	.5		154.1	64.7	87.8	12.1	9458	12940	.1	.15	.23	.1	.15	.27	.43	.33	.41
1300		49.0	1.8	4.4	.5		156.1	66.3	88.1	12.0	9589	13120	.07	.07	.2	.11	.11	.25	.3	.27	.27
1310	4	OFF Endurance																			
1315	2	ON Endurance																			
1330		68.5	2.0	6.0	2.5		152.7	61.2	90.4	11.9	9475	12980	.12	.05	.22	.11	.12	.3	.3	.27	.31
1345		67.8	2.0	5.5	2.6		155.6	63.4	90.4	12.1	9585	13130	.12	.07	.2	.1	.15	.27	.37	.29	.37
1400		68.0	2.0	5.5	2.5		155.5	63.0	90.4	12.2	9551	13090	.12	.07	.19	.1	.12	.27	.37	.27	.36
1415		68.0	2.0	5.5	2.5		154.9	62.8	90.4	12.2	9510	13030	.15	.07	.19	.11	.1	.26	.32	.3	.35
1430		68.4	2.0	5.5	2.5		154.2	62.1	90.0	12.2	9465	12940	.13	.07	.19	.11	.1	.26	.3	.3	.35
1445		67.2	2.0	5.2	2.7		156.6	63.7	91.6	12.2	9609	13160	.16	.07	.19	.11	.15	.25	.35	.35	.35
1448	2	OFF Endurance																			

Remarks 1000 AM 4 QTS, 1045 4 QTS, 1115 4 QTS, 1130 4 QTS, 1145 4 QTS | 1200 4 QTS, 1215 4 QTS, 1230 4 QTS, 1245 4 QTS, 1300 4 QTS | 1330 4 QTS, 1345 4 QTS, 1400 4 QTS, 1415 4 QTS, 1430 4 QTS | 1345 4 QTS

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Pratt & Whitney Aircraft U.A.

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 3/4   Date 2/10/78   Engineer Big Gamache   Operators [illegible]

D-4 Stand   Engine/Rig No. F-34024   Build 10   Project

Type of Test 50 HR ENDURANCE RUN

"A" Switch °F

Time A.M./P.M.	Test Idea (Mission Point No.)	T1 Oil Tank	T2 Oil Pump Dis.	T3 2/3 Comp Sep	T4 1-4-5 Oil Sup.	Ts1 2/3 Comp Air	Ts2 2/3 Comp Air	T6 1-4-5 Air Ori.	T71 Forw Cham Temp	T72 Forw Cham Temp	T81 Rear Cham Temp	T82 Rear Cham Temp	T9 Bore Temp	T10 Forw Dome Temp	T12 Forw Dome Temp	T11 Rear Dome	T13 Rear Dome	T15 Brea. Air Orif.	T16 2/3 Scav. Rim Disc	T14 Bore Air Out
		A-1	A-2	A-3	A-4	A-5	A-6	A-7	A-8	A-9	A-10	A-11	A-12	A-13	A-14	A-15	A-16	A-17	A-18	A-19
0915	2	55	51	50	51	57	57	56	56	56	55	55	50	57	56	55	54	57	56	57
0930	START ROTATION																			
1015		212	213	190	188	224	225	59	292	295	385	387	318	310	310	354	357	199	234	362
1035	4	ON ENDURANCE																		
1045		222	222	193	191	234	234	63	408	418	520	521	524	427	429	531	534	362	241	426
1100		218	219	198	197	238	240	66	404	413	551	552	547	453	453	568	570	217	257	461
1115		225	226	203	200	241	242	67	406	415	524	565	558	456	456	589	592	216	253	472
1130		232	233	212	210	250	252	69	406	415	561	562	563	455	455	595	599	223	255	473
1145		226	226	206	204	240	242	72	402	409	564	564	577	451	451	594	598	214	252	480
1200		224	225	200	198	237	238	71	401	408	568	568	583	449	450	596	[illegible]	213	245	485
1215		226	226	204	203	239	241	72	401	409	568	568	584	450	450	[illegible]	[illegible]	[illegible]	250	487
1230		238	239	216	214	251	253	72	406	411	560	561	579	[illegible]	454	567	[illegible]	[illegible]	[illegible]	477
1245		233	232	207	204	244	246	72	409	417	568	554	[illegible]	[illegible]	461	[illegible]	597	212	247	475
1300		234	232	200	198	243	243	73	405	412	546	546	[illegible]	[illegible]	[illegible]	[illegible]	577	217	240	[illegible]
1310	4	OFF ENDURANCE																		
1315	2	ON ENDURANCE																		
1330		236	238	218	[illegible]	[illegible]	[illegible]	74	471	484	562	564	576	549	549	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1345		235	225	201	199	248	249	75	458	470	598	598	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1400		230	231	200	197	246	246	75	455	467	601	601	597	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1415		227	228	199	195	243	243	76	446	458	[illegible]	[illegible]	[illegible]	510	510	[illegible]	636	[illegible]	[illegible]	[illegible]
1430		228	228	[illegible]	[illegible]	[illegible]	[illegible]	76	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	501	624	625	[illegible]	[illegible]	[illegible]
[illegible]		231	231	201	192	245	245	77	430	440	601	601	632	488	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
1445		OFF ENDURANCE																		

1500 Remarks STOP

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PWA F[illegible] REV 3 71

Pratt & Whitney Aircraft

LOG OF ENGINE TEST

EXPERIMENTAL TEST DEPARTMENT

Sheet No. 4/4
Date 2/1/78
Engineer Bill Gamble
Operators Conley/Rennick

D-4 Stand   Engine/Rig No. E34024   Build 10   Project

Type of Test 50 HR. ENDURANCE RUN

Time A.M. P.M.	Tot. Hrs	T17 Fwd Dome Air Orif A-10	T18 Rear Dome Air A-21	T19 G/B Oil In A-22	T20 G/B Oil Out A-23	B-11 No. 2 Brg. B-1	B-12 No. 2 Brg. B-2	B-21 No. 3 Brg. B-3	B-22 No. 3 Brg B-4	B-31 Pump Drive Brg. B-5	B-32 Pump Drive Brg. B-6	B-41 Pump Drive Low Brg B-7	B-42 Pump Drive Low Brg B-8	B-51 Low Tower Shaft Brg B-9	B-52 Low Tower Shaft Brg B-10	B-61 Upper Tower Shaft Brg B-11	B-62 Upper Tower Shaft Brg B-12	T21 G/B Brg #1 A24	T22 G/B Brg #2 A25
0915	2	50	52	55	54	57	/	57	/	53	50	55	56	57	[illegible]	58	58	55	55
0930		START ROTATION																	
1015		432	334	75	99	231		256		232	224	240	237	246	[illegible]	249	248	109	120
1035	4	ON ENDURANCE																	
1040		536	535	82	121	232		257		227	218	240	233	267	131	244	241	139	161
1100		466	583	86	129	257		280		257	249	264	260	282	147	264	261	147	170
1115		471	407	87	131	246		258		242	232	246	245	273	141	255	252	149	171
1130		472	537	87	131	251		263		247	239	249	249	277	[illegible]	259	255	149	172
1145		467	504	89	133	252		271		252	241	252	259	286	[illegible]	268	265	151	174
1200		463	490	90	134	240		257		236	228	245	243	272	139	252	249	151	174
1215		467	521	90	134	247		264		245	236	255	253	267	[illegible]	262	[illegible]	152	175
1230		474	564	89	128	256		275		254	246	263	261	289	151	268	[illegible]	[illegible]	[illegible]
1245		478	560	84	121	238		256		225	224	241	238	265	[illegible]	265	243	134	146
1300		469	551	85	116	231		242		223	216	228	226	256	[illegible]	236	[illegible]	128	130
1310	4	OFF ENDURANCE																	
1315	2	ON ENDURANCE																	
1330		593	683	94	116	301		309		312	281	276	[illegible]	319	[illegible]	303	296	131	146
1345		552	667	86	124	237		254		231	224	240	238	267	133	249	245	139	[illegible]
1400		540	651	89	125	240		258		235	227	242	241	269	[illegible]	250	247	[illegible]	152
1415		[illegible]	635	90	125	241		259		236	227	249	248	[illegible]		252	249	138	[illegible]
1430		572	634	89	126	234		257		235	227	243	241	277		251	247	[illegible]	[illegible]
[illegible]		504	623	88	129	[illegible]		262		234	230	247	245	274	[illegible]	255	[illegible]	[illegible]	[illegible]
[illegible]		OFF ENDURANCE																	

1500 Remarks STOP

Page No.   Report No.

PWA FRI ... REV 3/71   FD 4895

APPENDIX P
SYSTEM SAFETY ANALYSIS REPORT

This appendix contains the System Safety Analysis Report which documents the safety analysis performed during the design and fabrication of the test hardware. This report is included as a part of the final report as specified in paragraph 7.0 of the Statement of Work (Section F of Contract F33615-75-C-2075).

COMPARTMENTAL LUBRICATION SYSTEM PROGRAM

SYSTEM SAFETY ANALYSIS REPORT

Prepared Under Contract F33615-75-C-2075
For
Air Force Aero Propulsion Laboratory
Air Force Systems Command
United States Air Force
Wright-Patterson AFB, Ohio 45433

Prepared by D. R. Smith

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UNITED TECHNOLOGIES ®

COMPARTMENTAL LUBRICATION SYSTEM PROGRAM

SYSTEM SAFETY ANALYSIS REPORT

This System Safety Analysis Report identifies and describes the system and component malfunction modes of the system test rig (Rig No. F34024) for the Compartmental Lubrication System Program. System features and procedures which will be employed to prevent damage to the hardware or injury to test personnel are listed. This report satisfies the requirements of paragraph 7 of the contract (F33615-75-C-2075) and was conducted in accordance with paragraph 5.8.2.1 of MIL-STD-882.

Attachment A provides the Preliminary Hazard Analysis for the system rig hardware at both the system and component level. A brief description of each column of the Preliminary Hazard Analysis form is as follows:

Column 1. Hazard - This column lists the applicable malfunction mode(s) for the component. All recognized hazard modes for the component are listed and each is a basic condition analyzed in columns 2 through 6.

Column 2. Operation Phase - This column lists the operational phases in which a malfunction constitutes a hazard.

Column 3. Effect(s) - The effect(s) of the components abnormal condition on its operation is shown.

Column 4. Hazard Class - The hazard mode is classified in accordance with MIL-STD-882.

Class I - Negligible

Class II - Marginal

Class III - Critical

Class IV - Catastrophic

Column 5. Hazard Control - This column is used to list system features and/or procedures that may be employed to control hazardous conditions.

Column 6. Remarks - This column includes additional information needed to clarify or verify information in the other columns. Recommendations to improve system safety are also provided in this column.

Attachment B lists the precautions which are taken at the test facilities to prevent damage to the system rig and to prevent injury to test personnel. A flow schematic for the system rig is shown on Figure 1. System and component layout drawings used for this analysis are as follows:

L-232724	Compartmental Lubrication System Rig
L-231899	Compartmental Lubrication System Oil Tank
L-231893	Compartmental Lubrication System Oil Pump

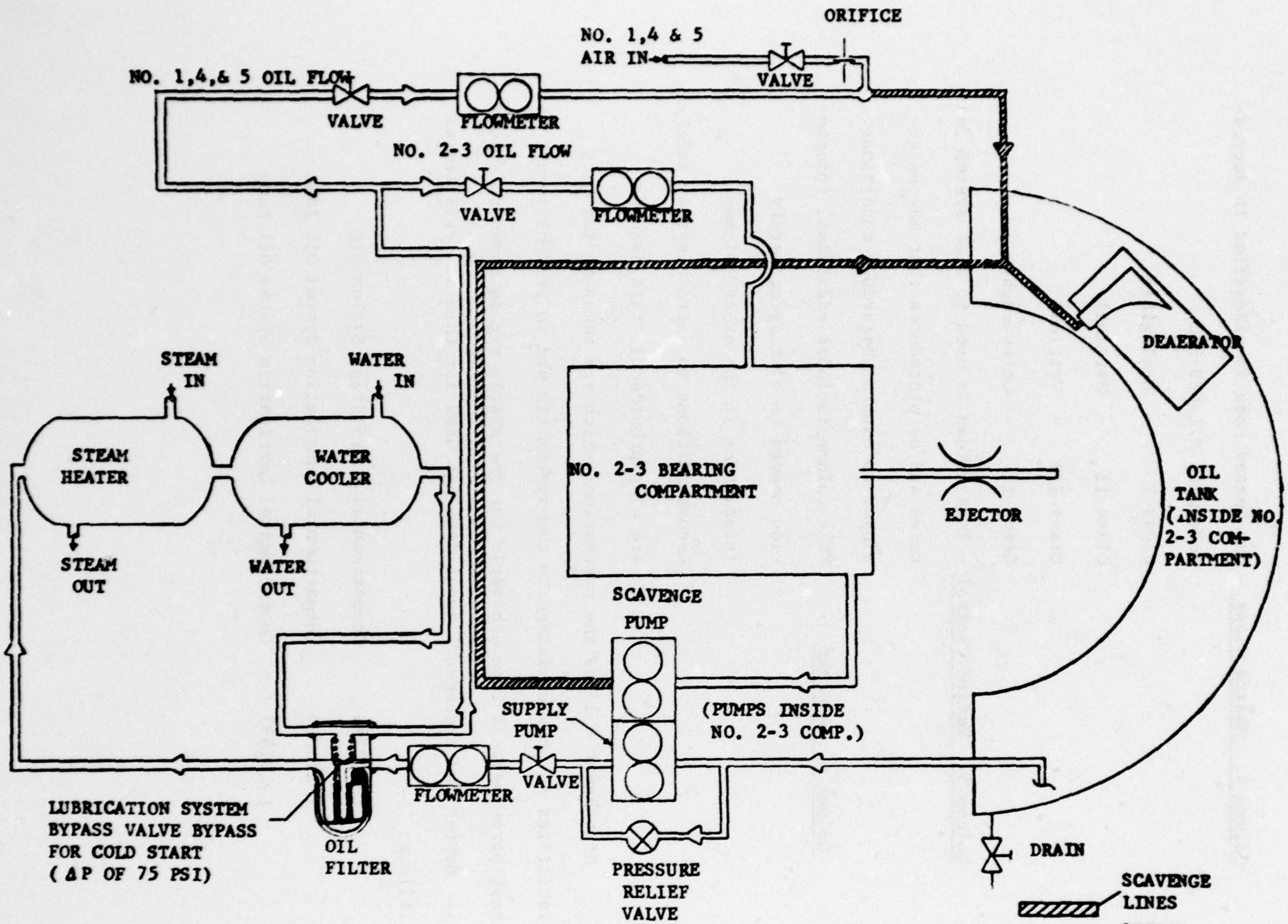

Figure 1. System Test Schematic

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ATTACHMENT A

PRELIMINARY HAZARD ANALYSIS

PREPARED BY: W. QUIGLEY   PG. 1 OF 6

REVIEWED BY: G. W. SCOTT   ISSUE DATE 7-30-77

ITEM COMPARTMENTAL LUBRICATION SYSTEM   SUBSYSTEM NA   SYSTEM NA

1 HAZARD	2 OPERATIONAL PHASE	3 EFFECT(S)	4 HAZARD CLASS	5 HAZARD CONTROL	6 REMARKS
System Analysis					
Leakage of rig covers, flanges and tube connections.	System test runs.	Degradation of lubrication system. Leakage of hot air and/or hot oil and possible fire hazard if conditions are suitable to initiate and sustain combustion.	III	• Seals on mating surfaces. • Leak checks and repair at assembly. • Flange fasteners and tube connectors incorporate locking features. • Rig instrumentation provides visual assessment of system temperatures, pressures and vibration, allowing operator to take appropriate action. • Rig stand fire suppression system.	
Rig Rotor Overspeed	System test runs.	Severe knife edge seal wear and disruption of thrust balance cavity pressures leading to possible compartment face seal and bearing distress, loss of lubrication and heat damage to rig parts. If overspeed condition is allowed to exceed safe limits uncontrolled rupture or rotor parts is likely and could result in a fire hazard.	III	• System instrumentation monitors rig rotor speed parameters. • Manual abort and visual monitoring capability is provided. • Isolation of personnel and shielding of rig compartment. • Rig stand fire suppression system.	

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ATTACHMENT A - PRELIMINARY HAZARD ANALYSIS

PREPARED BY: W. QUIGLEY PG. 2 OF 6
REVIEWED BY: G. W. SCOTT ISSUE DATE 7-30-77

ITEM COMPARTMENTAL LUBRICATION SYSTEM SUBSYSTEM NA SYSTEM NA

1 HAZARD	2 OPERATIONAL PHASE	3 EFFECT(S)	4 HAZARD CLASS	5 HAZARD CONTROL	6 REMARKS
Loss of bearing and gear lubrication	System test runs	Overheating of gears and bearings with resultant seizure of bearings could result in considerable damage to rig parts if rig test run is not aborted in time.	II	• System instrumentation monitors rig bearing condition. • System chip detector identifies incipient gear and bearing distress.	
Structural damage or gear train malfunction	System test runs	Degradation of lubrication system performance, excessive vibration misalignment of parts resulting in extensive damage to rig parts and fire hazard if conditions are suitable to initiate and sustain combustion.	III	• Rig design safety margins are in accordance with current PWA standard practice. • Stringent maintenance and inspection procedures. • Rig instrumentation provides visual assessment of system parameters allowing operator to take appropriate action. • Manual abort and visual monitoring capability is provided. • Isolation of personnel and shielding of rig compartment. • Rig stand fire suppression system.	
Contamination of lubrication system	System test runs	Degradation of lubrication system performance. Would require rig unscheduled shut-down, investigation, and repair.	II	• System filtration and chip detector • Stringent maintenance and inspection procedures. • System instrumentation monitors system performance.	Scheduled SOAP analysis is recommended to detect incipient distress.

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ATTACHMENT A - PRELIMINARY HAZARD ANALYSIS

PREPARED BY: W. QUIGLEY — PG. 3 OF 6

REVIEWED BY: G. W. SCOTT — ISSUE DATE 7-30-77

ITEM COMPARTMENTAL LUBRICATION SYSTEM — SUBSYSTEM NA — SYSTEM NA

1 HAZARD	2 OPERATIONAL PHASE	3 EFFECT(S)	4 HAZARD CLASS	5 HAZARD CONTROL	6 REMARKS
Rig Instrumentation Malfunction	System test runs	Erroneous parameters would be displayed causing rig unscheduled shut-down, investigation, and repair.	II	• Rig instrumentation provides visual assessment of system parameters allowing operator to take appropriate action.	
COMPONENT ANALYSIS					
Tank Assy - Lubricating Oil					
Dent, Chafe, Crack, or loose mounting and fittings	System test run	Oil leakage and flooding of #2-3 bearing compartment, reduced oil flow and increased part wear.	II	• Stringent maintenance and inspection procedures. • System instrumentation monitors system performance.	Borescope inspection access is recommended to allow visual inspection of oil tank.
Collapse of walls, seams, or loose servicing cap.	System test run	Oil leakage and flooding of #2-3 bearing compartment, reduced oil flow and possible oil starvation of main bearings and gear train.	II	• Same as above.	Same as above.

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ATTACHMENT A - PRELIMINARY HAZARD ANALYSIS

PREPARED BY: W. QUIGLEY PG. 4 OF 6
REVIEWED BY: G. W. SCOTT ISSUE DATE 7-30-77

ITEM COMPARTMENTAL LUBRICATION SYSTEM SUBSYSTEM NA SYSTEM NA

1 HAZARD	2 OPERATIONAL PHASE	3 EFFECT(S)	4 HAZARD CLASS	5 HAZARD CONTROL	6 REMARKS
Filter - Oil					
Cracks, loose fasteners, or partial loss of sealing	System test runs	Leakage of oil and unscheduled parts repair or replacement.	I	• Stringent maintenance and inspection procedures. • Filter leakage can be detected visually on rig stand.	
Clogged element	System test runs	Progressive reduction of oil supply and possible parts damage.	II	• Filter visual indicator button flags need for filter maintenance. • Filter is full flow non bypass with 70 micron metal wire mesh element.	
Oil cold start bypass valve fails to open	System test runs	Lack of oil flow downstream of main oil filter. Out-of-limit low oil pressure indication resulting in unscheduled rig shutdown and parts repair or replacement.	II	• Stringent maintenance and inspection procedures. • System instrumentation provides visual assessment of oil flow parameters.	Cold oil tests which would require bypass valve to open are not part of the system rig test schedule.

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ATTACHMENT A - PRELIMINARY HAZARD ANALYSIS

PREPARED BY: W. QUIGLEY | PG. 5 OF 6
REVIEWED BY: G. W. SCOTT | ISSUE DATE 7-30-77

ITEM COMPARTMENTAL LUBRICATION SYSTEM | SUBSYSTEM NA | SYSTEM NA

1 HAZARD	2 OPERATIONAL PHASE	3 EFFECT(S)	4 HAZARD CLASS	5 HAZARD CONTROL	6 REMARKS
Pump Assembly - Main Oil					
Score, erosion or parting surface seal leak	System test runs	Reduced and fluctuating oil supply pressure. Increased likelihood of gear and bearing distress, unscheduled repair or replacement of rig pump parts.	II	• Stringent maintenance and inspection procedures. • System chip detector indicates incipient distress. • System instrumentation monitors system pressure and flow parameters.	
Oil pump drive shaft malfunction	System test runs	Immediate stoppage of lubrication flow; overheating of gears and bearings with resultant seizure of bearings.	II	• System instrumentation monitors rig bearing condition. • System chip detector indicates incipient distress.	

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ATTACHMENT A-PRELIMINARY HAZARD ANALYSIS

PREPARED BY: W. QUIGLEY   PG. 6 OF 6
REVIEWED BY: G. W. SCOTT   ISSUE DATE 7-30-72

ITEM COMPARTMENTAL LUBRICATION SYSTEM   SUBSYSTEM NA   SYSTEM NA

1 HAZARD	2 OPERATIONAL PHASE	3 EFFECT(S)	4 HAZARD CLASS	5 HAZARD CONTROL	6 REMARKS
TUBE ASSEMBLIES					
Loose, nick, chafe or crack	System test runs	Light leakage of oil and possible minor oil pressure fluctuation. Increased oil service and repair/replacement of parts.	I	• Visual inspections. • Tubing supports in adequate numbers maintain natural frequency between supports.	
Fracture, perforation or disconnect	System test runs	Leakage of oil and flooding of #2-3 bearing compartment if internal, reduced oil flow and increased part wear.	II	• Stringent maintenance and inspection procedures.	
		External leakage will result in reduced oil flow and increased part wear.	II	• System instrumentation monitors system pressure and flow parameters. • Visual inspection.	
Blockage of flow passage	System test runs	Possible partial to complete oil supply starvation resulting in damage to gears and bearings. Would require rig unscheduled shut-down investigation and repair.	II	• Stringent maintenance and inspection procedures. • System instrumentation monitors system pressure and flow parameters.	

ATTACHMENT B

COMPARTMENTAL LUBRICATION SYSTEM TEST FACILITY SAFETY REVIEW

1. The rig area is weather protected by a roof and is open on two sides, with restricted access.
2. Stand personnel are inside an air conditioned control room separated from the rig area by an 8" concrete wall, containing a blast resistant viewing window.
3. Test area piping systems are designed in accordance with the American national standard code for pressure piping, ANSI B31.3-1973, "Petroleum Refinery Piping, Division A."
4. Test area tubing systems are designed in accordance with MIL-F-5509-C, "Military Spec. Fittings, Flared Tube, Fluid Connection."
5. Valves, flanges and gaskets are designed in accordance with ANSI B16.5, "Steep Pipe Flanges and Flanged Fittings."
6. Fire protection is provided by three separate systems.
   A. Water spray fixed system, designed in accordance with NFPA No. 15, which can be activated manually by stand personnel.
   B. Dry powder "Ansul" system <u>for fire inside the test chamber</u>, activated manually by stand personnel.
   C. Stand personnel can also activate a steam system for fire control when conditions require additional protection.
7. The rotating drive systems are protected by over-speed sensors. Lubricating oil low level warning, over-temperature warning systems, variable speed coupling water level indicator and water pressure warning signals. A water outlet over temperature sensor will shut down the drive motor.

8. The rotating drive systems are equipped with vibration indicators which are monitored by stand personnel inside the control room.

9. Piping, tubing and pressure vessels are protected by ASME approved pressure relief valves set to relieve at 10% above the system operating pressures. These valves dump to safe disposal systems.

10. Electrical installation is in accordance with NFPA 70, the National Electric Code.

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